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NOTES ON THE METALLOGRAPHY OF THE FERROUS METALS.

BY WIRT TASSIN.

These notes originated as a series of talks to Foundry and Mill Chemists and to Engineers of Tests. In gathering the data no attempt was made to cite references or to give individual credit. This condition was due in part to the informality of the talks and because there was then no intention of publishing the notes. They contain little original data.—W. T.

Metallography deals with the internal structure of metals, which may best be studied with the microscope.

Metals are opaque, their structures can be seen only with reflected light, that is by surface illumination, which may be either "oblique" or "vertical."

With the former, which is used chiefly for low-power work the light is directed obliquely upon the surface of the metal, as in Fig. 1, Plate 1. A part of the light is reflected away from the tube of the microscope and a part passes up through it to the eye, thus causing certain portions of the field to appear in relief. This method of illumination has only a limited application.

APPARATUS FOR PHOTO-MICROGRAPHY.

Vertical illumination is used for all powers and is the only one suitable for high-power work. With this method, use is made of a device known as a "vertical illuminator," which may be of the prism or, preferably, of the mirror type. The latter is represented by the "Beck vertical illuminator," Fig. 2.

A metal container which holds a thin, circular, microscope cover glass is mounted on a support, *d*. The body of the container is pierced by one or more openings of varying diameters and can be rotated completely about the collar *c*. The collar is tapped externally with a thread, and screws into the lower end of the tube of the microscope. The other end, *a*, of the container is threaded on the inside to take the nose piece and the objective. Turning the plate, *d*, to some desired angle reflects the illuminating ray, *L*, which enters the aperture, *e*, down upon the surface of the metal. This in turn reflects the light up through the tube to the eye, Fig. 3.

The effect produced by an oblique illumination is shown in Fig. 4 and that from a vertical illumination in Fig. 5, Plate II. These are micrographs of wrought iron.

Some special device is usually necessary to fully illuminate the surface of the metal and to get sufficient light through the microscope to the eye.

An apparatus designed and used by the writer for this purpose is shown in Fig. 6. This instrument is compact, readily adjusted, and can be attached to any microscope having a society thread. It possesses the advantage of having the source of light always traveling with the barrel of the microscope, thus permitting the use of any power and of any range of focus without disturbing the alignment of the illuminating apparatus. It also obviates the necessity of providing the stage with a rack-and-pinion movement so necessary with other illuminating devices.

The apparatus consists of an arm which locks into the microscope tube by means of a threaded collar. The arm serves to carry the condensing arrangement, consisting of a telescope tube about one inch in diameter, which carries the lenses. The tube has a maximum extension of four

inches and a minimum of two, thus allowing for a wide range of focus. The telescope tube is mounted in a sleeve provided with a set screw to lock it in any position. A hanger from the arm is attached to this sleeve by means of a trunnion controlled by a set screw, thus permitting the tube as a whole to be tilted at any angle. A vertical adjustment is also provided for by means of a set screw at the outer extremity of the arm. The rear of the telescope tube carries a shield which is provided with

Fig. 6.—TASSIN ILLUMINATOR WITH MICROSCOPE AND CAMERA.

clips to carry any desired color screen. Fastened to the shield is a rod on which there is a movable carrier for the source of light. This consists of two pieces of tubing, brazed together at right angles. Each of the pieces is provided with a set screw so that the source of light may be moved up or down, front or back, or turned on its axis and set at any angle with reference to the plane of the rear lens of the condensing train. This permits the use of any or all of the light.

The source of light may be an acetylene jet, one of which is furnished with the apparatus, or it may be an electric lamp. The electric light is a No. 68 6-volt, 16-c.p., 12-G. (Headlight), candelabra base, tungsten lamp, which may be purchased through any electrical supply house. It is mounted as in Fig. 7. The socket may be obtained by breaking the ordinary porcelain candelabra base and using the brass. The support is of thick-walled brass tubing 3/16, or 0.187 of an inch in diameter. One end is threaded to take a screw which binds the socket to the tube by means of the two fiber washers and at the same time gives the bottom contact for the lamp.

Any pattern of microscope may be used, but the description here given refers to the Bausch and Lomb B.B.H. type. The microscope is without a substage, and has had the pillar raised one inch by putting a brass washer between it and the stage. Fastened to the top of the barrel, from which the draw tube has been removed, is an "adapter," which fits a "light-tight connector" carried on the lens board of the camera. Fitting centrally in the "adapter" is a brass sleeve centered by a collar and carrying the eye piece. The details of these devices are seen in Fig. 8. Screwed into the collar of the illuminating device as attached to the microscope is a "Beck vertical illuminator." Below this is a "quick-acting nose piece," which permits of a ready change of objectives without any unscrewing motion.

The apparatus as described lends itself readily to either vertical or oblique illumination. It permits the use and ready interchange of any eye piece and objective and may be immediately attached to a camera. The illumination, when all is in proper adjustment, is brilliant and uniform and is adapted both for visual examinations and for the making of photographic records.

A convenient outfit for general use consists of: *B.B.H. microscope* without the Abbe condenser and screw substage. and with the pillar raised as suggested. *The sleeve, adapter, and connector.* The *Beck vertical illuminator. 7.5x eye piece.* 10.0x eye piece. 32 m.m. objective. *16 m.m. objective. 8 m.m. objective.* 4 m.m. objective. 1.9 m.m. objective. Quick-acting nose piece with rings. Mechanical stage. Eye piece or stage micrometer. *Prest-o-lite tank, motorcycle size.* All of these items are desirable and those in italics are essential.

A photographic camera is a necessity. Any camera having a shutter may be made to serve on making a stand for it by simply fastening two pieces of board together at right angles.

The boards should be of one-inch stuff, about eight inches wide. The vertical one should be about thirty-six inches long, and, starting about four inches from the top, a half-inch slot should be cut down the center to within twelve inches of the bottom. This slot is the channel for a set screw which serves to lock the camera in any desired place, up or down. The bottom board should be about twelve inches long and should be weighted with lead to insure stability. It is better, of course, to use some special form of camera, fixed on a stand so that it is possible to swing the camera out of the way to render the microscope quickly accessible for eye observations. Such a camera is seen in Fig. 6.

In using the camera it is always necessary to re-focus the microscope after the visual examination has been made, so that a sharp image may be obtained on the ground glass. The best results are obtained by cementing a large microscope cover glass on the under side of the ground glass with Canada balsam, thus making a clear spot. The focus is now fixed with the coarse and fine adjustment of the microscope, and the sharpness of the image as seen on the clear spot determined with the aid of a pocket magnifier, using the fine adjustment for the close focusing. The size of the image is regulated by racking the bellows. It is generally better to cover the entire ground glass with the image and mat out the parts not desired when printing from the negative.

The magnification is measured by placing on the stage a micrometer scale or any other measure, bringing the scale divisions into focus and measuring the distance between any two of the divisional lines as seen on the ground glass. The magnification is this distance divided by the actual distance between the lines on the scale. It will be found advisable to plot the magnifications obtained for the different tube and bellows' lengths and to refer them to a distance scale marked on the upright of the camera, so that the magnification may be read at once for a given bellows length with a given eye piece and objective.

In making the photographs any of the plates used should

be fast and should give excellent color values. The time of exposure is necessarily a variant, and can be determined only by experience; it ranges from 15 seconds to 15 minutes. After exposure the plates are developed in the ordinary way. It is suggested that hydroquinon or metol be used in place of "pyro" as a developer. The first two give white negatives; the last, especially when tank developed, gives a yellowish negative. For tank development "Ingento edinol-quinol developing tablets" are recommended.

PREPARATION OF THE SPECIMEN.

The development of the structure in a metal is in principle simple, though somewhat complicated in its details. It is first necessary to polish it to a mirror finish, and if this be carefully done it is possible to show the structure by this means alone (Fig. 9, Plate II). The development of structure solely by polishing is tedious, and in general the results are more or less unsatisfactory, so that it is usually expedient to stop polishing when a fair surface has been obtained and then to bring out the structure by attacking the surface with some solvent. This will show the structure with clearness as illustrated by Fig. 10, which is the same section as seen in Fig. 9, etched with ammonium persulphate.

To prepare a surface for a metallographic examination, the metal is cut or broken to some convenient size. The sample is then ground or filed flat on one face. Most metals yield readily to the file, and cross filing followed by draw filing gives excellent surfaces when properly carried out. When filing, the use of flat, six-inch Grobet files, numbered from 0 (coarse) to 6 (dead smooth), is recommended. The files are to be used in their numerical order, first in one direction and then in a direction at right angles to this when the next finer file is used, care being taken that the marks made by one filing are completely obliterated by the succeeding one. A drop of oil should be used with each filing and both the specimen and the file kept clean. It will usually be found advantageous to true the surface after filing by rubbing it on a sheet of glass

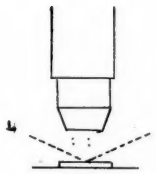


Fig. 1

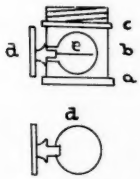


Fig. 2

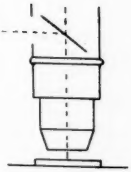


Fig. 3

Fig. 7

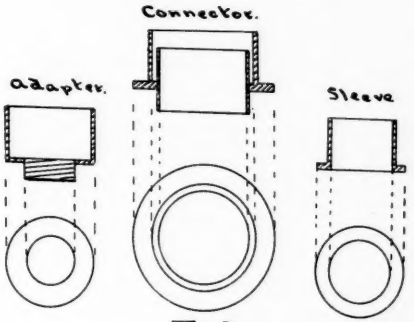


Fig. 8

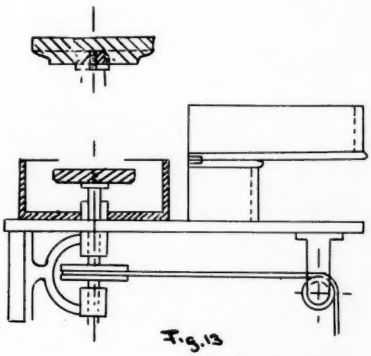


Fig. 13

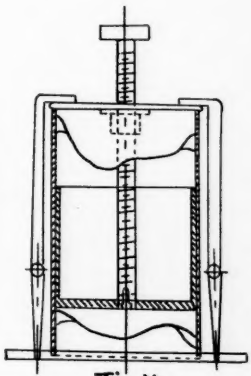


Fig. 16

Plate I.

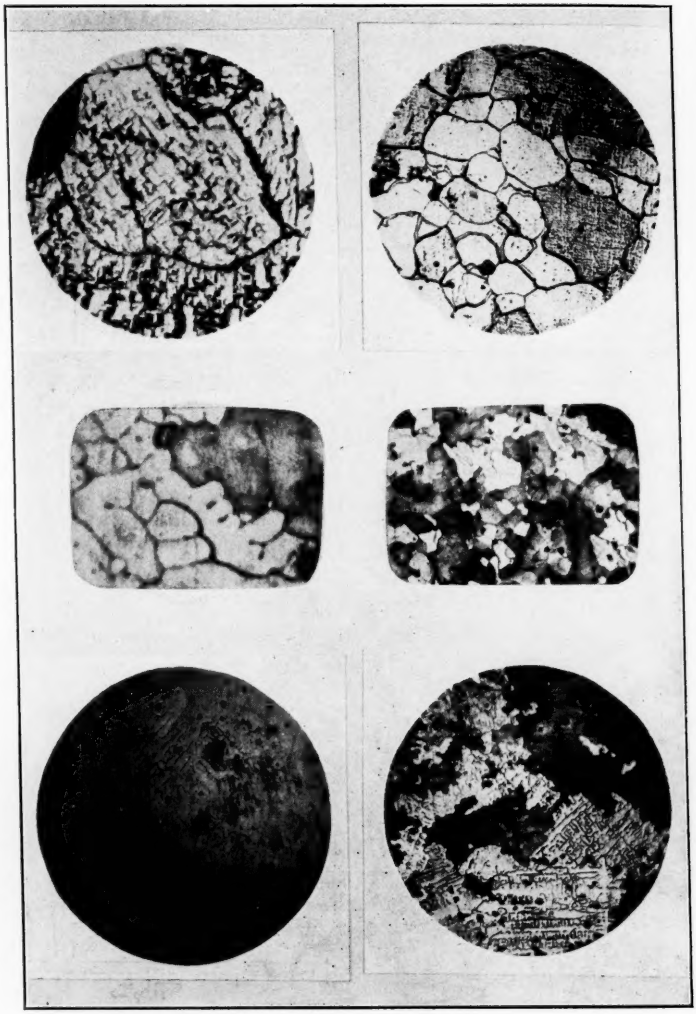


FIG. 4.—Oblique illumination.
FIG. 14.—Copper etched with nitric acid.
FIG. 9.—Copper polished in relief, × 150.

FIG. 5.—Vertical illumination.
FIG. 15.—Copper etched with ammonium persulphate.
FIG. 10.—Copper etched with ammonium persulphate, × 150.

Plate II.

with a little "washed flour emery" moistened with a little water.

When a metal is too hard to file, a flat face may be ground by means of an emery wheel. This should be followed by a rubbing on glass with water and flour emery. When a metal is very soft a file clogs and cannot be used, then a flat surface may be obtained by means of a sharp paring or cutting tool, as a knife or a wood chisel. This should be followed by a rubbing on the glass plate with rouge or rotten stone and water; emery should not be used.

Often a specimen is so small that it cannot readily be held in the fingers. A convenient vise may be made by tapping a thread in the center hole of a "fast-joint butt hinge," fitting a screw therein and thus drawing the leaves of the hinge down on the section and holding it firmly in one position during the filing and subsequent operations.

Very thin sections, such as sheet metal, may be soldered on supports made of short lengths of half-inch brass tubing. The solder used should have a low melting point, so that the heat of soldering does not alter the structure of the metal. Guthries eutectic has a very low melting point and consists of: lead, 20.25; tin, 21.10; bismuth, 50; cadmium, 12.50. Lipowitz's alloy melts at 149 degrees F., and is made of lead, 26; tin, 12.78; bismuth, 50; and cadmium, 10.40. Another alloy that melts at 155 degrees F., contains lead, 1; tin, 1; bismuth, 4; and cadmium, 1. Another solder is made from lead, 1; tin, 2; and bismuth, 1.

POLISHING THE SPECIMEN.

After a flat surface has been obtained the specimen is ready for polishing. This operation in itself is simple, yet at the same time there is often room for the display of considerable skill in giving the final surface. Too much care cannot be exercised in keeping the polishing powders, blocks, or heads clean. A small particle of a coarser powder falling on a finer one, or even a bit of dirt, may produce a series of deep scratches that will necessitate re-filing. Therefore each wheel, or block, and

each abrasive should be kept in a covered box by itself. After each operation the specimen should be thoroughly cleaned with water and alcohol to remove the least trace of the abrasive. The polishing may be done either by hand or by power.

With hand polishing, the specimen, after being rubbed on the glass plate with washed flour emery and water and washed, is rubbed on pieces of cloth fastened to blocks of wood. The first block has stretched upon it a piece of piano felt charged with the finest emery knife polish or with black magnetic rouge moistened with oil. The second block is covered with a piece of the finest wash leather and is charged dry with a small amount of the best gold rouge. The specimen is rubbed on the first block till all the marks left by the flour emery are removed. It is then washed with water and alcohol and the process repeated on the second block till a polish is obtained, the washings following as a matter of course.

With power polishing use is made of the machine shown in Fig. 13, Plate I.

This is motor driven and is so geared that the heads travel at about 2,000 revolutions. The construction is very simple and, exclusive of the motor, can be made in any shop at a small cost. It consists of a sewing-machine frame from which the treadle and flywheel have been removed.*

Across the top of the stand is bolted a wooden table top about one inch thick and of an area slightly greater than that of the stand. The top carries the bearing and hanger for the drive and spindle, one end of which is threaded to take the heads. These are made of hard wood and carry on the bottom face a brass plate which is tapped to take the thread on the spindle of the drive. Fitting somewhat closely to the rim of the head is an iron ring, conveniently made from a section of a 6-inch pipe, which holds firmly in place the piece of cloth that carries the abrasive. In the sketch is seen a removable wooden box which forms a guard to the wheel while polishing. One of these should be provided for each wheel, and the No. 1 box used only with the No. 1 wheel, and so on.

For ordinary work but four heads are used, each of which is kept in its own box. Two of these heads are covered with 10-ounce duck, one with felt made by the American Felt Co., of New York, and known as style 1234, No. 1936, weight 31¼; the fourth is covered with either velvet or broad cloth. The first canvas-covered wheel is charged with "Flour

*To adapt a sewing machine to foot power the treadle and flywheel should be left on the stand. A countershaft is installed with a grooved pulley over the flywheel end and a bevel gear at the other to engage a gear which replaces the grooved pulley on the spindle in the sketch.

emery paste," the second with "Washed flour emery paste." The felt-covered head is charged with "Gold and silver rouge F." The velvet wheel is used for finishing.

The specimen is given as high a finish as possible on the No. 1 wheel and is washed thoroughly in water and then in alcohol. It is then put on the No. 2 wheel, and so on, care being taken to wash thoroughly after each grinding.

DEVELOPING THE STRUCTURE.

After polishing, the metal is ready to have its structure developed. This is done usually by means of some reagent which attacks or colors some constituents of the metal more than it does others. The character of the etch figure is dependent upon the kind of reagent used and the manner in which it is applied, therefore there is at this point room for the display of considerable skill and ingenuity. To illustrate this, compare the structure of a piece of copper etched with nitric acid, Fig. 14, Plate II, with a similar piece of copper etched with ammonium persulphate, Fig. 15.

The structure of a metal may also be developed by taking advantage of the fact that in many metals some constituents are differently colored from the others when the whole is heated. The polished specimen is heated gradually on an iron plate to some desired temperature, usually above 100 degrees C., in air, or in air containing traces of hydrogen sulphide, vapor of iodine, chlorine, bromine, etc., followed by a quick cooling by floating on mercury. The effect of heat tinting is well brought out by the heating of a phosphatic steel; the pearlite becomes blue, the cementite red and the phosphide yellow. In developing structure it should be remembered that with very soft metals the operation of polishing may cause a surface drag which will give a false structure on etching. Here a system of alternate polishing and etching will develop the true structure. The surface is etched, re-polished with as little pressure as possible, again etched, and so on till the true structure appears.

After a proper etching and drying the specimen is now ready for mounting.

Use is here made of a device, Fig. 16, Plate I, which consists of two telescoping pieces of brass tubing of any convenient diameter (the one figured is four inches) ground to fit. The outer tube carries across its top, and securely fastened to it, a brass strip which is centrally tapped to receive a screw with a milled head, which is about an inch longer than the outer tube. This adjusting screw is fixed to the inner tube by means of a screw bottomed in it in such a manner that the inner tube may be raised or lowered but cannot revolve. The inner tube should be of a length sufficient to give a good bearing and not be thrown off of level. The lower end is closed with a face plate as indicated. The whole is set up on a base plate provided with a groove which receives the outer tube and which also carries two hinged clasps screwed into the plate and provided with set screws to lock the tube firmly when in position.

The method of use is as follows: A lump of modeling wax is stuck on any slide suitable for the purpose. These are easily made by cutting pieces of window glass into rectangles of varying sizes as 1 inch $\times$ 2 inches, 2 inches $\times$ 3 inches, etc., and the specimen set up in the proper position on the wax. The slide is then placed on the base plate, the tube placed in the groove and the whole clamped down. The specimen is leveled perfectly by turning the milled head of the adjusting screw so as to bring the bottom of the inner tube, which must as a mechanical necessity travel evenly, down on the specimen, forcing it into the wax, thus giving a serviceable mount and assuring a level surface regardless of the shape of the sample. The inner tube is now lifted slightly by means of the adjusting screw, the set screws released, the clamps thrown back, the tube lifted off and the specimen is ready for study.

THE STRUCTURE OF METALS.

A metal is made up of small, irregular grains which result from the process of solidification. Solidification of the metal begins at a number of different centers, each of which probably originated as a single minute crystal or crystallite and grew by the aggregation of a number of similar crystals, all, or the majority of which, are in parallel position so far as any one group is concerned, but not necessarily so with regard to any other group. The rate of growth is probably a variable, and differs in different directions, and is apparently limited only by

contact with another group, so that each forms an irregularly shaped granule, hence the granular structure. The slower and more uninterrupted the rate of cooling the larger will be the growth of the grains and the more symmetrically will they arrange themselves in relation to each other. The growth of the grain is due to the continued deposition of crystallites or minute crystals in close contact. Given the proper conditions, as the grain increases in size it becomes more compact, and any foreign matter which may separate the crystals is squeezed out into the channels between the faces of the grains. This squeezing out process may be arrested by pressure or by quick cooling.

A metal which naturally forms large crystals is generally weaker than one whose crystalline structure is small, especially when much impurity is present: first, because it is more prone to continue the crystalline growth at a lower temperature; second, because the impurities tend to arrange themselves along crystal and grain faces and prevent perfect contact, these faces then become planes of weakness and fracture occurs along them. When a metal is subjected to pressure both the grain and the individual crystals are distorted and the arrangement of the whole is disturbed. When the pressure is carried far enough a state of unequal equilibrium is set up, the metal becomes weak, rigid, brittle and is easily fractured. The equilibrium may in instances be re-established by heat.

A "virgin metal" is composed of a single chemical element in a reasonable state of purity.

An "alloy" is a mixture of two or more metals or metallic substances.

ALLOYS.

Alloys are solutions, liquid or solid, according to their physical condition. A solution is a chemically and physically homogeneous mixture whose composition may, within certain definite limits, undergo a continuous variation. The limits of this variation are the limits of the existence of the solution.

An alloy of two metals in either the liquid or solid state may be regarded as: A solution of one metal in another. A chemical combination. A mechanical mixture. A solution or a mixture of any two or all of the foregoing. The different constituents of an alloy may all be united in the liquid state and yet separate on cooling, forming two or more alloys of different densities and different fusibilities. Also many solid alloys release certain of their constituents when heated, and these constituents have different freezing points from the remainder of the alloy.

In the cooling of certain fused mixed metals a certain alloy of these metals freezes out and the most fusible alloy, called the " eutectic," is left. The addition of a third metallic solvent to an alloy, two of whose constituents do not naturally mix, often causes their assimilation by destroying the eutectic. When an alloy of two metals yields neither definite combinations nor isomorphic mixtures there is only one eutectic When two metals yield one or more definite combinations there are two or more eutectics. When two metals form isomorphic mixtures there is no eutectic. If the crystalline structure of an alloy appears most often on a fractured surface, or if it is easy to develop by polishing and etching, it is probably not a case of eutectic alloys or of isomorphic mixtures, since these latter have a structure of extreme fineness and give to a fractured surface an appearance like glass. The effect of heat on an alloy containing a eutectic is to increase the influence of that eutectic. The lower the melting point of any element which forms a eutectic with a metal the lower will be the melting point of that eutectic. The greater the number of foreign elements in a metal, the lower will be the melting points of the eutectics. The lower the melting point of a eutectic the better are the conditions for the production of a large granular structure. The larger the structure the more brittle is the metal.

A typical section of a pure metal will show when polished and etched a number of irregular polygonal areas outlined by fine lines, as in Fig. 17, Plate III. That of an alloy made up

Fig. 17 Fig. 18 Fig. 19 Fig. 20 Fig. 21

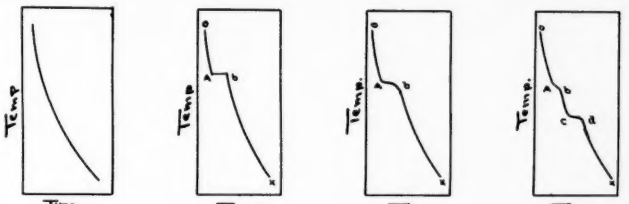


Fig. 22 Fig. 23 Fig. 24 Fig. 25

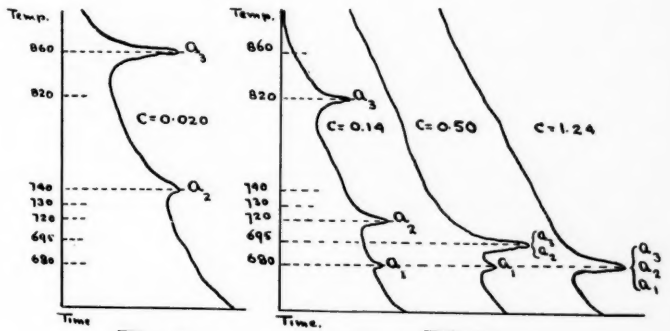


Fig. 26 Fig. 27

Plate III.

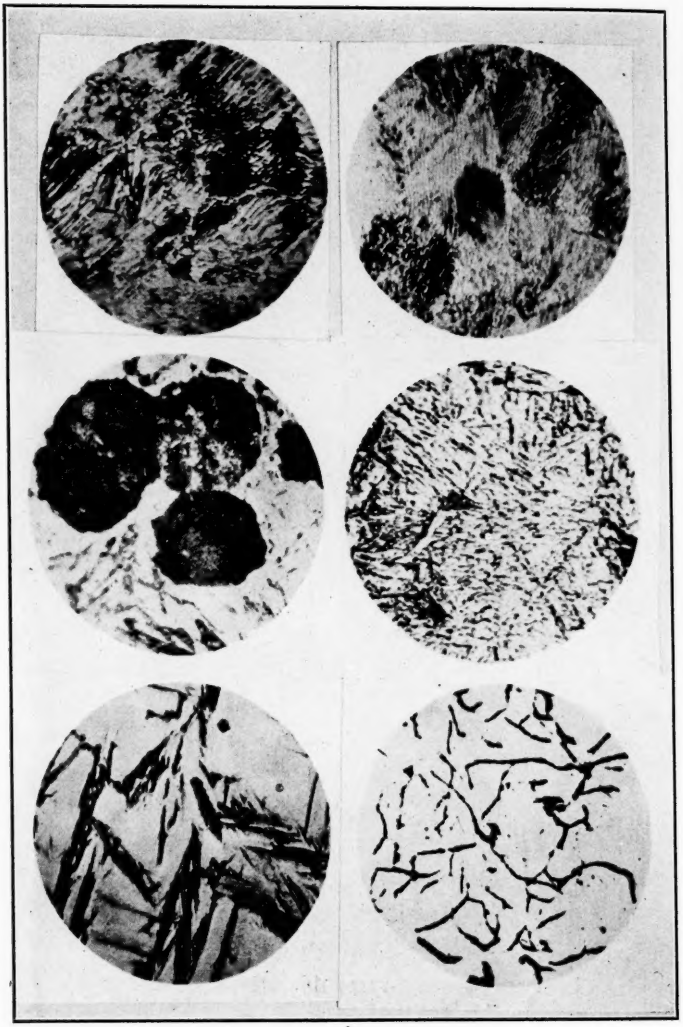


FIG. 30.—Pearlite.
FIG. 31.—Sorbite, dark spot, with pearlite.
FIG. 32.—Troostite, dark spots, with martensite.
FIG. 33.—Martensite.
FIG. 34.—Austenite (light) with hardenite in 1.5 carbon steel (Osmond).
FIG. 35.—Graphite in cast iron.

Plate IV.

of two metals will show when polished and etched a structure similar to that of a pure metal if the components are completely soluble in each other when solid, *i. e.*, if they form a solid solution. A section of an alloy of two metals, which when solid are completely insoluble in each other, will show one of several forms of structure which is dependent upon the relations the component metals may bear to each other. The structure may be made up of a greater or less proportion of one pure metal in a matrix of the other pure metal, as in Fig. 18, in which the primary crystallizations of the first are seen surrounded by a secondary crystallization of the second. Or the alloy may have a structure as in Fig. 19, in which the mixture shows primary crystallizations of the first surrounded by a eutectic of the two. In Fig. 20 the primary crystallizations are of the second metal with the eutectic. Or the structure may consist entirely of the eutectic, as in Fig. 21.

THE EFFECT OF TEMPERATURE ON METALS.

It is now in order to consider briefly the effect of temperature on metals and to examine their cooling curves, since by this method we have a simple means of determining whether or not there is any drastic change with the temperature in the mass and the point at which this change, if any, takes place.

If the temperature of a virgin metal cooling from below its melting point be taken at regular intervals and its values be plotted on a chart so that the vertical distances represent temperature and the horizontal ones time, the curve will be perfectly even. (Fig. 22, Plate III.) This simple curve shows that there has been no change in the nature of the metal cooling under these conditions. Now take the cooling curve of the same metal in the same way, but from the liquid to the solid state. The slope of the curve falls with the temperature till a certain point is reached when the fall is arrested and the temperature remains stationary for a time, finally the curve retakes its original slope and there is again the simple curve. Examining the curve, Fig. 23, we find that there is a marked

break at the line *ab* and that the corner *b* is quite sharp, the significance of this will appear later. The change in the curve as shown by the break is indicative of a change in the metal, and we know that physically it has changed from the liquid to the solid state.

Now add a small amount of a second metal which is soluble in the liquid first metal and trace the cooling curve. The new curve, Fig. 24, differs from the former in that there is a distinct rounding of the corner *b* in the line *ab*, and in general there is a lowering of the freezing point. This indicates that the solid which here separated was not a pure metal but a solid solution of the second in the first. Now greatly increase the quantity of the second metal so that it exceeds the amount which can be held dissolved in the first when in the solid state. The cooling curve, Fig. 25, again shows the rounded corner at *b*, but there is a second halt in the curve which begins abruptly at *c* and continues to *d*, where the previous slope is abruptly regained. At *c* there have been two solids deposited alternately from a homogeneous liquid with the temperature remaining constant till the whole mass has solidified. A section of this alloy will show alternate lamellae of the two metals, in other words, a eutectic. Any change, therefore, in the physical or chemical nature of a metal or an alloy during its cooling or its heating will show itself in its temperature curve.

The heating and cooling phenomena of iron and its alloys furnish a typical illustration of the above statement. During the gradual heating of the ferrous metals there is a retardation and even a reversal in the rise of temperature. The expansion is checked and reversed so that the metal contracts and then re-expands. There is a change in the specific gravity. The thermo-electric deportment becomes anomalous. The electrical conductivity varies and the magnetic values alter. Finally, there are astonishing changes in the specific heat. Further, most, if not all, of these changes occur simultaneously.

When a piece of carbonless iron is heated a slower rise of temperature is noted at 740 degrees C. and another at 860 degrees C., Fig. 26, Plate III. These lags in the temperature

rise indicate an absorption of heat. In cooling there are analogous phenomena—at these same temperatures there are delays in the fall of the temperature. These indicate an emission of heat.

ALLOTROPIC FORMS OF IRON.

These anomalies are explained on the assumption that iron assumes at least three allotropic states and changes successively from one to the other according to temperature conditions. Iron below 740 C. is in the condition known as "Alpha." It crystallizes in cubic forms, is not hard, is tough, malleable and strongly magnetic. Alpha iron has little or no capacity for dissolving carbon. Iron between 740 and 860 C. is said to be in the "Beta" state. Like alpha iron, it is cubic, and may form isomorphic mixtures with it; also it has but little affinity for carbon. Unlike alpha iron, it is hard and non-magnetic. Iron above 860 C. is said to be in the "Gamma" state. It is soft, non-magnetic, and has a marked affinity for carbon.

The temperatures at which these allotropic changes occur are known as "critical temperatures" or "points." The region near 740 C. is called Ac_2 for heating and Ar_2 for cooling, that near 860 C. is called Ac_3 and Ar_3 respectively. The temperature Ac is always higher for the same sample than the corresponding Ar. The difference is due to hysteresis and may amount to several tens of degrees. When iron contains carbon its first critical point is in the region of 680 C., known as A_1. Here there is during the heating a marked absorption, and during cooling a quick emission of heat. The point A_1 remains practically a constant, regardless of the carbon content of the metal. The point A_3, which in carbonless iron is at 860 C., falls rapidly as the carbon increases until it blends with A_2. This, in turn, under the above conditions, begins to fall until it begins to blend with A_1, Fig. 27.

These conditions may be explained as follows: The ferro carbon alloy at temperatures below 680 C. contains carbon combined as the carbide of iron, Fe_3C. At 680 the carbide separates into carbon and iron, and the amount of heat re-

quired for this separation accounts for the lag in the temperature. Reciprocally, in cooling, beginning at the region above 680, the carbide re-forms, and its heat of formation causes the rise in temperature. With the increase of the amount of carbon dissolved in gamma iron the point A_3 falls. The point A_2, that of the formation of beta iron, which does not dissolve carbon, remains constant at 740 C. until A_3, in falling, reaches it. But as the quantity of carbon increases, gamma iron is formed in ever-increasing quantities and at ever-lower temperatures until the production of beta iron is entirely eliminated and gamma iron at 740 is converted to alpha iron.

When a molten solution of iron having 4 per cent. of carbon cools to ordinary temperatures, there is first, when solidification begins, a mixture of a molten solution and a solid solution of carbon in iron. The mass on further cooling solidifies in a 2 per cent. solution of carbon and iron with free carbon and gamma iron. As the temperature falls some of the free carbon combines with the gamma iron to form a compound, at the same time the 2 per cent. solution of carbon in iron breaks up to form other compounds, assumed to be sub-carbides of iron. On further cooling the sub-carbide breaks up to form a eutectic mixture of an iron carbide with iron. The final product is a crystalline mass containing iron, a carbide of iron, a eutectic of an iron carbide and iron, and free carbon.

Each of these bodies has been named, and they will now be considered individually with reference to their occurrence and their appearance under the microscope. These bodies are:

Ferrite.—This is more or less pure alpha iron, in that it may contain in addition to iron those elements which form isomorphous mixtures or solid solutions with it, as silicon, manganese, nickel, etc. Ferrite occurs in iron and steel, having less than 0.85 per cent. of carbon. It is rare in cast iron and is then found chiefly in the very silicious gray irons. Under the microscope it is seen as polyhedric grains, Fig. 28, Plate V, which are made up of small cubes. Ferrite is developed by

the action of nitric acid, both strong and dilute; by tincture of iodine; by a 5 per cent. solution of picric acid in alcohol, and by an 8 per cent. watery solution of cupric ammonium chloride. A good reagent is a 4 per cent. solution of nitric acid in amyl alcohol. Ferrite is soft and is readily scratched with a needle. Silicon greatly increases the size of the grain and of the cubes composing it. Chromium greatly decreases the size of the grain and of the individual crystals composing it.

It has been stated previously that alpha iron, or ferrite, has no solvent action on carbon. It is, however, probable that when a steel contains more than 0.50 per cent. of carbon this is dissolved by the ferrite up to about 0.27 per cent., but only in the presence of certain isomorphous mixtures which are comprised under the name of ferrite.

Cementite.—Carbide of iron, Fe_3C, containing 6.67 per cent. of carbon, Fig. 29, Plate V. It is very hard, 6.5 on the Mohs scale, or nearly the hardness of quartz, which usually causes it to stand out in relief on polishing. It may be separated chemically from the other constituents of steel by means of cold dilute hydrochloric acid. Cementite is not colored by any of the usual etching mediums and remains tin white. It is colored red after 30 minutes in a cold 2 per cent. solution of ammonium oxalate; and brown after 5 to 10 minutes in a boiling solution of 2 grams of picric acid in 24.5 grams NaOH in 73.5 grams of water. It is rarely found alone in steels containing less than 0.90 per cent. of carbon, but it is a characteristic of high-carbon steels and of white cast iron. Cementite forms a eutectic with ferrite, and in this form is found in all irons which contain carbon. It is isomorphous with the carbide of manganese, and for small amounts of this element the mixture is still called cementite. It is destroyed at 680 C. and re-forms on cooling. It may also be formed by the moderate heating of the compounds which contain it in solution.

Pearlite.—This is a eutectic, and, like all eutectics, it is made up of alternate layers of different substances. It is composed of alternate layers of ferrite and cementite, which are rarely more than one thousandth of a millimeter in thickness. Pearl-

ite appears under the microscope in laminated comb-like, or sheaf-like forms, and may occasionally be granular, Figs. 30, Plate IV, and 30A and B, Plate V. With an oblique illumination it often shows a pearly iridescence, hence the name. Pearlite is colored by nitric and picric acid solutions and by iodine. It is found in cast and annealed steels and in cast iron. It contains 0.85 per cent. of carbon, and hence a steel with this percentage of carbon must consist entirely of pearlite. It disappears when heated above 680 C., as a result of the separation of the cementite, and re-forms on cooling, but at a slightly lower temperature.

Sorbite.—This is a form of pearlite developed when steel is quickly cooled through the first change point, 680 C., without, however, being quenched, Fig. 31, Plate IV. Under these conditions a partial separation of the eutectic may occur so that the alternate layers of the pearlite cannot be distinguished; in other words, the swiftness of cooling is such that the cementite and ferrite do not have time to separate into layers of appreciable thickness. Sorbite is strongly colored by a 5 per cent. solution of picric acid in alcohol. It is hard and quite difficult to break, and is found in 0.85 carbon steel and in tempered steels heated between 200 and 300 degrees.

Troostite.—Like sorbite, this is an intermediate form and lies between pearlite and martensite, Fig. 32. It is formed when steel is quenched near the first critical point, or given a mild quenching, as with oil, from higher temperatures. It is regarded as a solution of carbon and alpha iron (see *ante* under "Ferrite"), and occurs as black spots in the midst of pearlite areas. Troostite is colored to the exclusion of all other constituents by a mixture of one part amyl, one part ethyl, and one of methyl alcohol in a 4 per cent. solution of nitric acid in acetic anhydride. Alcoholic solutions of picric or nitric acid also color it, as does a solution of one part of hydrochloric acid in 100 parts of ethyl alcohol. This last does not affect martensite.

Troosto-Sorbite closely resembles, and differs from troostite only in the manner of its formation. It is formed at a

high temperature in a solution of carbon in gamma iron, which becomes alpha iron on cooling. Troostite is a solution of carbon in alpha iron formed directly at a re-heating temperature. Troosto-sorbite appears as irregular black-edged patches, generally showing cementite towards the center and which are therefore less dark in that region.

Osmondite.—This is intermediary between troostite and sorbite. It is a solution of carbon formed under special conditions as in "hammer hardening."

Martensite.—This is the very hard substance of which hardened steels consist and is characteristic of steel hardened under ordinary conditions, *i. e.*, at a heat slightly over 680 C. and quenched in water, Fig. 33. It appears as interlacing needles or fibers roughly arranged in triangular shapes. Martensite is a solution of cementite in an isomorphous mixture of alpha and beta iron.

Hardenite.—This is a form of martensite containing 0.80 per cent. carbon, that is, a carbon-saturated martensite.

Austenite.—This is gamma iron having carbon in solution. Fig. 34. It is found associated with hardenite and martensite in steels containing 1.60 to 1.80 per cent. of carbon which have been heated above 1,000 C. and quenched in brines cooled below 0 C.

Graphite.—This appears in iron alloys containing 6 per cent. of carbon, or when silicon is present, at a much lower percentage. It is easily recognized by its color and curved shape. Fig. 35.

Temper Carbon.—This is the finely-divided carbon which separates from cast iron during a prolonged annealing, as in the process of malleablizing. It may be a graphitic carbon that has separated as a result of the decomposition of cementite by prolonged heating above 900 C., or it may be the original graphite of the iron which has been diffused or sublimed as such. It occurs in small, more or less rounded, grains.

Ferrite and pearlite are found in all classes of steel used in structural work, in rails, etc. Cementite is found in all high-carbon steels. Martensite is characteristic of all steels hard-

ened by quenching. Sorbite is sometimes found in steels that have been quenched after rolling. Troostite, osmondite and austenite are rarely met with. Graphite is a normal constituent of cast iron. Temper carbon is found in malleable cast iron.

EFFECT OF OTHER ELEMENTS ON IRON.

The ferrous metals also contain small amounts of other substances which have an important influence on the quality of the product. They are—

Silicon.—As an element silicon is soluble in all proportions in iron and dissolves in the ferrite without altering its character except to form larger and more perfect crystals, Fig. 36, Plate V, and to increase the grain size. Aside from this it cannot be distinguished under the microscope. Combined as silicate it is a characteristic component of wrought iron (*q.v.*) and is not infrequent in steel, Fig. 37.

Sulphur.—Occurs as the sulphide of iron and of manganese. As the former, it is found chiefly in steels very low in manganese and under these conditions tends to form mesh-like structures enveloping the grain, Fig. 38. As sulphide of manganese it segregates in quite large patches, Fig. 39.

Phosphorus.—Forms a series of compounds and a solid solution with iron. It may be dissolved up to 1.7 per cent. as a solid solution, and in this form it cannot be recognized under the microscope, unless an increased grain size can be regarded as evidence. When the amount of phosphorus is larger, a eutectic containing 10.2 per cent. of phosphorus is formed, Fig. 40, Plate VI. Under certain conditions a definite phosphide of iron, Fe_3P. is formed, Fig. 41. This compound is tin white in color, and occurs in lath-shaped forms and platy layers. Cupric ammonium chloride is an excellent etching medium and may be used for quantitative separation, as the phosphide is insoluble in it. Heat tinting colors the compounds of phosphorus yellow. The compounds of phosphorus possess a tendency to segregate. Further, in the presence of carbon the amount of phosphorus that can be held in solution

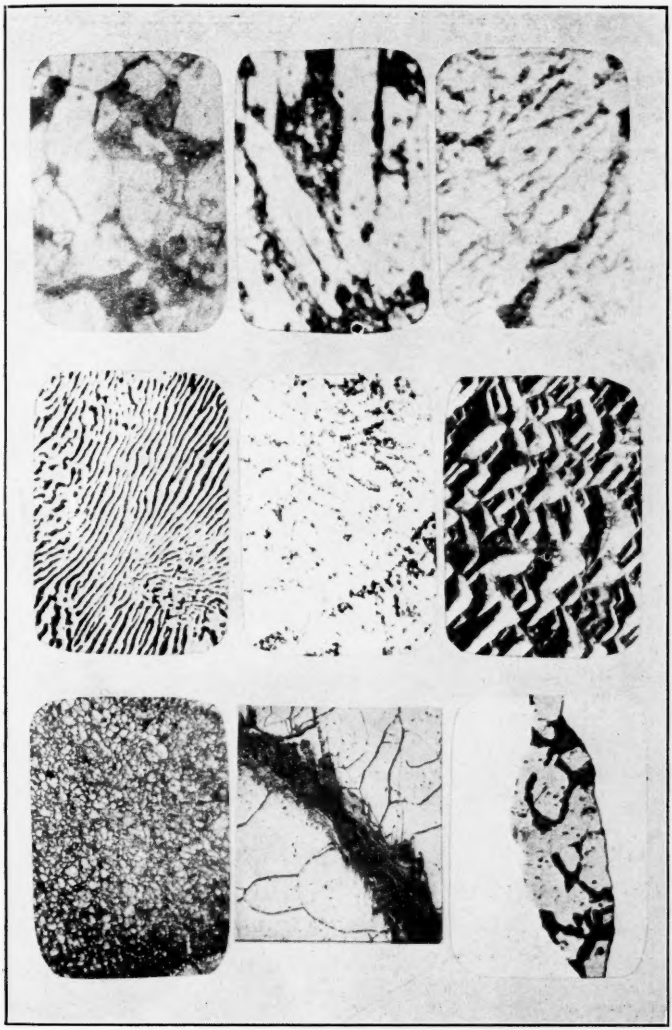


FIG. 28.—Ferrite.
FIG. 30A.—Lamellar pearlite (Osmond).
FIG. 38.—Sulphide of iron enveloping the grain.
FIG. 29A.—Cementite as etched with nitric acid.
FIG. 30B.—Granular pearlite (Heyn).
FIG. 37.—Slag.
FIG. 29B.—Cementite as etched with ammonium oxalate.
FIG. 36.—Four per cent. silicon steel showing increased crystal size (Stead).
FIG. 39.—Sulphide o manganese with slag.

Plate V.

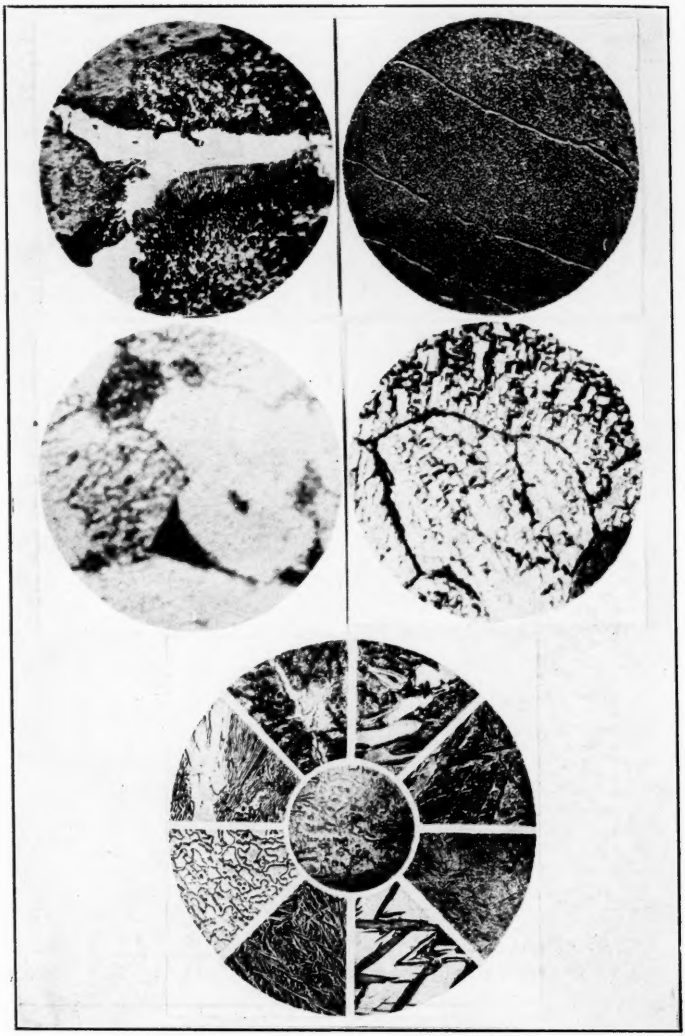


FIG. 40.—Fringe of phospho-carbide eutectic in cast iron.
FIG. 41.—Phosphide of iron etched with cupro-ammonium chloride.
FIG. 42.—Nearly carbonless iron.
FIG. 43.—Cubic crystals of iron.
FIG. 50.—Influence of heat treatment on the structure of 1.5 carbon steel (Austen).

Plate VI.

by the iron decreases as the carbon increases, hence the higher the carbon the more pronounced is the influence of phosphorus on the metal.

Manganese, Copper, Nickel, Chromium, etc.—In general these elements dissolve in the iron to form manganese, copper, nickel, chromium ferrites, cementite, etc.

The influence of these several elements on the physical properties may be summarized as follows:

Tenacity is lowered by elements which tend to promote the formation of graphitic carbon, as silicon, phosphorus and sulphur; by elements which tend to increase the size of the grain, as phosphorus and silicon; and by elements which induce the formation of compounds which tend to separate the grain, as sulphur or phosphorus, and silicon in the form of slag, or oxygen as oxides. Tenacity is increased by elements which in themselves possess this property to a high degree, as nickel, cobalt and copper; by definite amounts of elements which hinder the separation of graphitic carbon, as manganese, chromium, titanium, etc., by small amounts of elements which act as deoxidizers and degasifiers and which hinder the formation of blow holes, as silicon, aluminum, manganese, titanium and vanadium; and by elements which tend to reduce the size of the grain, as nickel and chromium.

THE STRUCTURE OF IRON.

The word "Iron" will be taken to mean the product of the bloomery or that obtained by the boiling or puddling of pig iron. The metal so obtained is known commercially as "wrought iron." It may frequently contain over 99.75 per cent. of iron, and in general carries from 0.01 to 0.10 per cent. of carbon, together with a certain amount of slag present as a result of the process of manufacture.

In studying the structure of the purest of the wrought irons there may occasionally be found areas which are practically free from carbon and from impurities. Such a region will exhibit the structure of a pure iron, Fig. 42, Plate VI. This

structure is made up of grains of ferrite, which in turn are made up of cubic crystals, Fig. 43. Some of these grains appear quite bright, others are dark. This difference in appearance is due to the fact that the minute crystals which make up the individual grains are similarly oriented in the same grain, but differently in different grains, thus causing the several grains to reflect the light differently.

The structure of a longitudinal section of a piece of wrought iron is made up of grains, which in turn are made up of cubic crystals similar to those in pure iron. In addition there are numerous slag areas which appear as irregular black lines having the same general direction, but which vary in thickness and in length. These lines run in the direction of the rolling, and to them is due the fibrous appearance characteristic of wrought iron when fractured. A cross section will show the same general structure, except that the slag, instead of appearing as fibers, now appears as irregular patches or blebs. In both sections there will be found between the ferrite grains a cementing material. This is pearlite, and in it will be found the whole of the carbon present in the metal.

Iron, up to a certain point, becomes stronger and tougher by repeated forging. The structure becomes finer, the slag threads thinner, and the grain is closer, smaller, and more uniform. A fine structure is generally an indication of strength, and in the examination of wrought iron attention should be paid to the size, uniformity and closeness of the grain as indicating the amount of work and the finishing temperature. The amount of slag is also a factor, since it prevents the cohesion of the grains between which it lies and is inversely proportional to the strength of the metal.

THE STRUCTURE AND PHYSICAL PROPERTIES OF STEEL.

The word "Steel" will be taken to mean the product of the cementation process, or the malleable compounds of iron made in the crucible, the converter, or the open hearth. The metal, regardless of its origin, records to a certain extent its history

in its structure, and steels of a given kind possess certain characteristics in common. The structure and physical properties of a steel may be profoundly modified by the rate of cooling from a high temperature, the degree of re-heating, and by the amount and kind of work to which it has been subjected. For example, in a steel obtained by the simple solidification of the liquid metal followed by a slow and undisturbed cooling the grain size will be large. If now the metal be re-heated to some temperature below its melting point and cooled slowly without mechanical work there will be a corresponding change in the appearance of the grain. If it be worked while hot there will be a further change, and if after cooling it be worked cold there will be a still different appearance. These changes are illustrated by the micrographs of a mild steel which show the structure of the metal as cast, Fig. 45, Plate VII; as annealed, Fig. 46; as hot worked to a 6-inch round, Fig. 47, and to a 3/8-inch round, Fig. 48, and then cold worked to a No. 2 wire, Fig. 49. The effect of heat treatment has been beautifully brought out by Roberts-Austen, who has arranged the structures obtained at different temperatures for the same piece of steel as seen in Fig. 50, Plate VI.

The structure of steel, as well as its other physical properties, as tenacity, hardness, etc., depend upon—

A.—Its chemical composition.

B.—The heat treatment through which it has passed.

C.—The amount and kind of mechanical work to which it has been subjected.

THE CHEMICAL COMPOSITION OF STEEL.

The chemical composition of a steel influences both its physical properties and its structure.

In any steel, in addition to the iron, there is carbon, manganese, sulphur, phosphorus, silicon, etc. These elements form definite compounds which affect both the structure and the physical properties of the metal. Probably they have the following formulae: Fe_3C, Mn_3C, MnS, Fe_3P, FeSi, etc. Know-

ing the formulae and having the analysis of the steel, it is possible to calculate the percentage amounts of these compounds present.

Assuming a steel to have this composition: C, 0.58; Mn, 0.65; S, 0.05; P, 0.04; Si, 0.10; Fe (by difference), 98.58.

The 0.10 per cent. of Si will form 0.300 per cent. FeSi.

The 0.04 per cent. of P will form 0.257 per cent. of Fe_3P.

The 0.05 per cent. of S will form 0.135 per cent. of MnS.

The MnS contains 0.085 per cent. of manganese; subtracting this from the total Mn content, 0.65 — 0.085, leaves 0.565 per cent. to combine with the carbon to form 0.587 per cent. Mn_3C.

The carbide of manganese contains 0.022 per cent. of the total carbon. Then 0.58 — 0.022 leaves 0.558 per cent. of carbon to combine with the iron to form 8.37 per cent. of Fe_3C.

The proximate structural composition is then: Fe_3C, 8.37 per cent.; Mn_3C, 0.587 per cent.; MnS, 0.135 per cent.; Fe_3P, 0.257 per cent.; FeSi, 0.300 per cent.; Fe, 90.351 per cent.

These formulae are deduced by the methods commonly used in chemical computations. The equation representing the reaction is first written, then the atomic weights of the substances entering into the equation and the result obtained by a simple proportion. Thus:

S + Mn = MnS. The atomic weights are: 32 + 55 = 87. It follows, then, that 32 S requires 55 Mn to make MnS. Therefore 0.05 per cent. of S will require $\frac{55 \times 0.05}{32}$ or 0.085 per cent. of Mn to make MnS. Also 0.05 + 0.085 is 0.135, the per cent. of MnS required.

To calculate the amount of pearlite and excess ferrite or excess cementite in steel containing a given percentage of carbon, assuming that the steel has been cooled slowly, so that everything is in equilibrium, the following apply:

In steels with less than 0.85 per cent. C.—The per cent. of cementite in pearlite is the per cent. of carbon in the steel multiplied by 15. The per cent. of ferrite in pearlite is the per cent. of cementite times 6.4. The per cent. of pearlite equals the per cent. of ferrite plus the per cent. of cementite. The per cent. of excess ferrite equals 100 minus the per cent. of pearlite. The total ferrite equals the per cent. of excess ferrite plus the per cent. of ferrite in pearlite.

Steels with 0.85 per cent. C.—The per cent. of cementite in pearlite is the per cent. of the carbon in the steel multiplied by

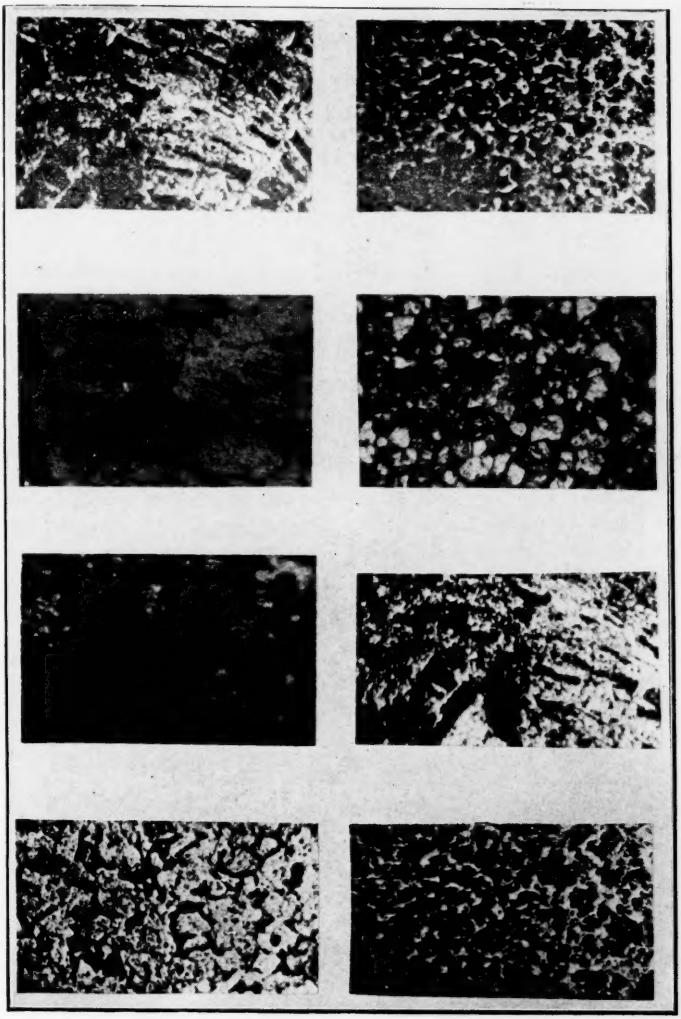


FIG. 45.—Steel as cast, × 60.
FIG. 46.—Steel as annealed, × 60.
FIG. 47.—Steel as hot worked to a 6" round, × 150.
FIG. 48.—Steel as hot worked to a ⅜" round, × 150.
FIG. 49.—Steel as cold worked to a No. 2 wire, × 200.
FIG. 51.—Steel, grain size, as cast.
FIG. 52.—Steel, grain size, with too high annealing.
FIG. 53.—Steel, grain size, with proper annealing.

Plate VII.

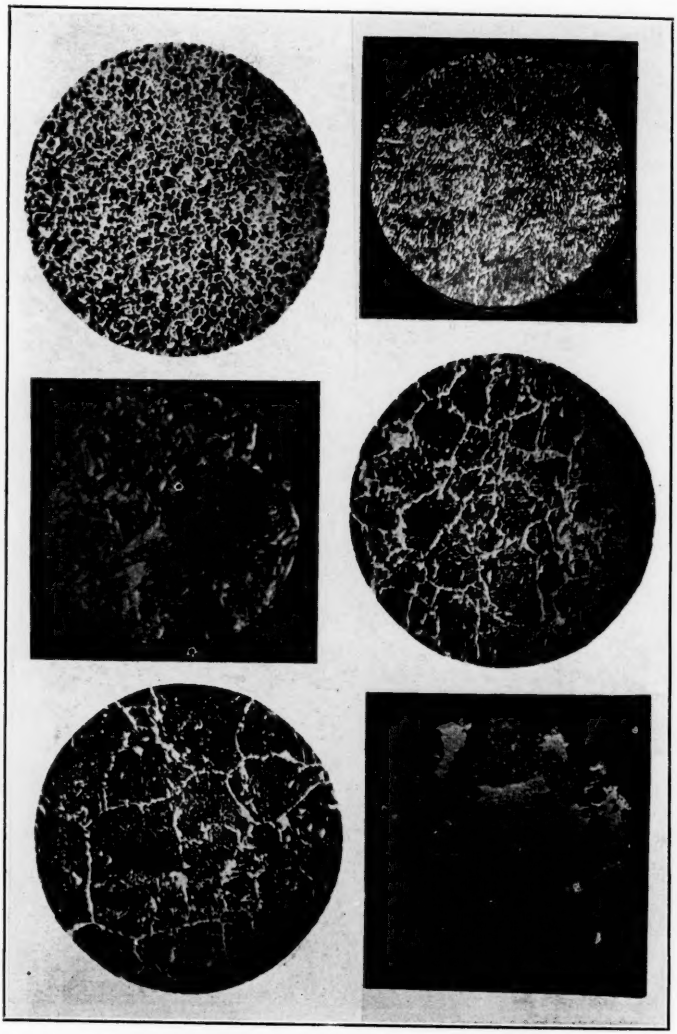


FIG. 54.—Steel, grain size at 900.
FIG. 66.—Steel at too low a hardening heat.
FIG. 56.—Steel, grain size at 1,300.
FIG. 65.—Steel at the proper hardening heat.
FIG. 55.—Steel, grain size at 1,200.
FIG. 67.—Steel at too high a hardening heat.

Plate VIII.

15. The per cent. of ferrite in pearlite is the per cent. of cementite times 6.4.

Steels with more than 0.85 per cent. C.—The per cent. of total cementite is the per cent. of the carbon in the steel times 15. The per cent. of ferrite in pearlite equals 100 minus the per cent. of total cementite. The per cent. of cementite in pearlite is the per cent. of ferrite divided by 6.4. The per cent. of pearlite is the per cent. of cementite plus the per cent. of ferrite. The per cent. of cementite in excess is the per cent. of the total cementite less the per cent. in pearlite.

Upon the distribution of these constituents, ferrite, cementite, pearlite, etc., and their relative proportions is dependent the physical properties of the steel, due allowance being made for the influence of heat treatment and work.

THE HEAT TREATMENT OF STEEL.

The heat treatment of steel includes the whole of the thermal conditions through which the metal has passed. Steel is heated to give the plasticity required for rolling or forging. It is heated to relieve internal stress. It is heated to give hardness or to take it away.

Annealing is the re-heating of steel to some given temperature for some given period of time and then allowing it to cool slowly, preferably out of contact with air and in some neutral atmosphere. The general effect of a proper annealing on the structure is to make the grain finer and more uniform. Heating for a longer or a shorter time at too high a temperature, or a long continued heating at too low a temperature produces a coarse structure, Figs. 51, 52, 53, Plate VII. The smaller the grain size, in general, the better the metal, and the higher the temperature from which steel cools, the larger the grain, Figs. 54, 55, 56, Plate VIII.

These conditions are typical and they have been generalized as follows:

A.—The higher the temperature above the first critical point, A_1, from which steel cools, the larger the size of the grain.

While if the A_1 point is the highest temperature reached, the steel will have the finest structure possible for it to assume.

NOTE.—In normal steel, when rolled, the size of the grain is inversely proportional to the sectional area of the bar.

B.—If hardened or unhardened steel be heated to the A_1 point all previous structure is obliterated and is replaced by the finest structure which the metal can assume.

NOTE.—This applies in detail to hard steels only, with mild and medium steels the old structure is not quite obliterated.

C.—If steel be heated for a time above 1,000 C. it acquires a coarse structure and becomes brittle.

These generalizations and their relations to the physical properties are illustrated by the following experiments upon a steel containing C., 0.330; Si, 0.023; Mn, 0.321, P, 0.029; S, 0.026. The A_1 point was at 675 C.

The tests were made on 12-inch lengths taken from a coil rolled to a ⅜-inch round. All sections were heated out of contact with air, held for two minutes at the temperature indicated, and then buried in lime till cold. The results given are the average of six measurements.

Temperature. Degrees C.	Yield point.	Red. area. Per cent.	Tensile strength.	Elong. in 4" Per cent.	Plate IX, Fig.
600	43,670	50	71,400	27	57
675	44,900	55	72,980	28	58
700	42,990	53	69,700	23	59
800	44,600	52	69,550	23	60
900	44,590	53	70,000	27	61
1,000	39,990	43	68,420	21	62

THE HARDENING AND TEMPERING OF STEEL.

Hardening results from the heating of steel to some temperature above A_1 and then quickly cooling it. This is accomplished by heating the steel to some certain temperature fixed by its chemical composition and then quenching it in water or oil, or in certain cases in a blast of air. The use

of air is restricted to a rather limited group of steels, and in practice water or oil are the more commonly used.

Water has a high specific heat and is very mobile. Its hardening effect is dependent upon the temperature of the bath, and the lower the temperature the greater the effect, so that water at 60 degrees F. will make steel harder than it will at 212 degrees F. Oil has a lower specific heat than water and is less mobile. Its quenching effect, at least as regards fish, sperm and linseed oils, is not influenced by its temperature, and the hardening effect is the same at any temperature between 32 and 250 degrees F. In general, water quenching is more efficient than oil quenching. The effect of each upon the structure is shown in Figs. 63 and 64, Plate IX.

The change from the soft to the hard condition takes place within a comparatively short range of temperature. This range is fixed by the critical point of the metal. Further, the more rapid the rate of cooling through this range the harder will be the metal. The rate of cooling above or below the critical range has little influence upon the hardness, but the maximum temperature reached and the time it is held there, while having only a minor effect upon the hardness, does have an effect upon the structure and hence upon the physical properties, resiliency, elastic limit and tensile strength. The effect of different hardening heats on the same steel, quenched in the same medium, is seen in Figs. 65, 66 and 67, Plate VIII.

Tempering is modifying the hardness by re-heating to a temperature below A_1 followed by a rapid or a slow cooling. The desired temperature may be obtained by means of heated baths of oil, lead and lead alloys, salt, etc., or by a simple re-heating. The degree of temper is usually judged by the color (called temper color) given to the surface of the metal as a result of oxidation. By varying the conditions of the heat treatment almost any degree of hardness can be produced. The effect of tempering is to cause the loss of a certain amount of hardness, to reduce brittleness, and to increase the toughness and ductility.

The effect of tempering on the microstructure of hardened

steel is to give a finer and more compact grain. It changes at 150 C. such austenite as may be present to sorbite. At 300 C. much of the martensite is decomposed. Above 300 C. the troostite and sorbite go over to cementite and ferrite, either as such or as the eutectic, pearlite.

THE WORKING OF STEEL.

It has been stated that the physical properties of steel depend upon the chemical composition, the heat treatment through which it has passed, and the amount and kind of work to which it has been subjected. It remains to consider now the influence of mechanical work.

Work may be applied in two ways, hot or cold. Its effects may be generalized as follows:

A.—If the work is vigorous enough to penetrate all parts of the mass no crystallization takes place during that work.

B.—Hot work as such has no direct effect upon the structure of the steel, but as it retards the critical change till a lower temperature is reached it may then influence structure.

C.—Cold work distorts the grain and flattens or elongates it in the direction of the work. Its influence is more pronounced than hot work.

It follows, then, that the lower the temperature at which the last effective work is applied the smaller will be the grain size. If the finishing temperature be too low the metal will be more or less fibrous, and if the work be carried far enough the metal will have a tendency to split along the direction of the work or to become brittle. If the finishing temperature be too high the coarser will be the grain and the weaker the metal. Hot working when finished at the point A_1 gives a small grain and the most tough and ductile metal. Hot working below a visible redness, that is, "blue working," causes a marked loss in ductility and a marked increase in brittleness. Its effect on the grain may be likened to that resulting from an exaggerated cold working. Cold working up to a certain point increases the tensile strength, elastic limit, stiffness and

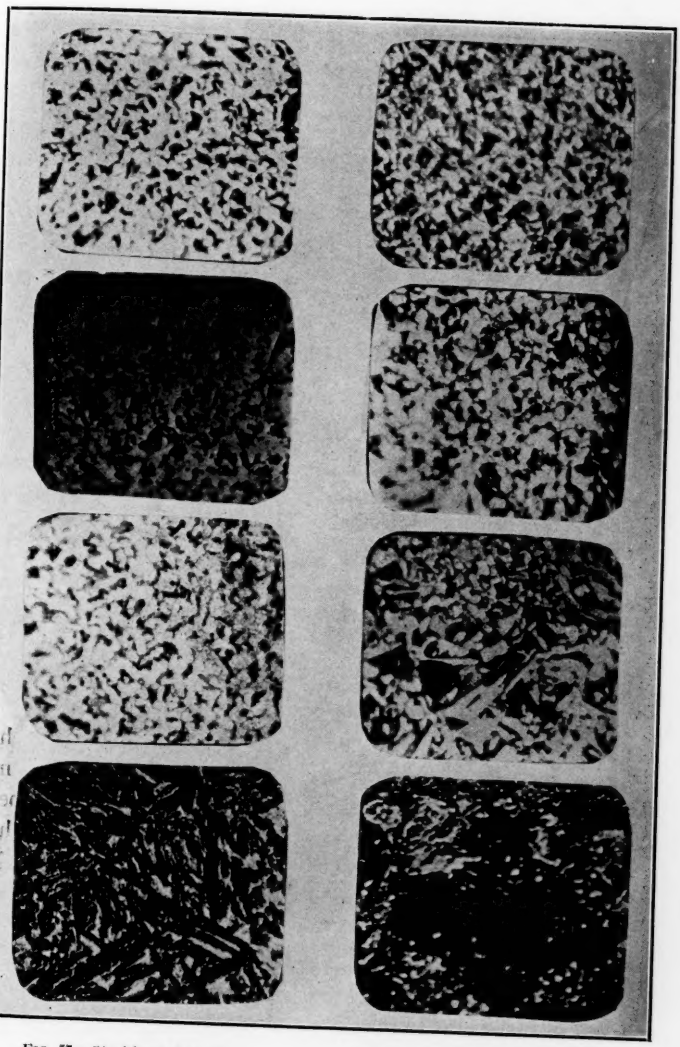


FIG. 57.—Steel heated to 600.
FIG. 58.—Steel heated to 675.
FIG. 59.—Steel heated to 700.
FIG. 60.—Steel heated to 800.
FIG. 61.—Steel heated to 900.
FIG. 62.—Steel heated to 1,000.
FIG. 63.—Steel as water quenched.
FIG. 64.—Steel as oil quenched.

Plate IX.

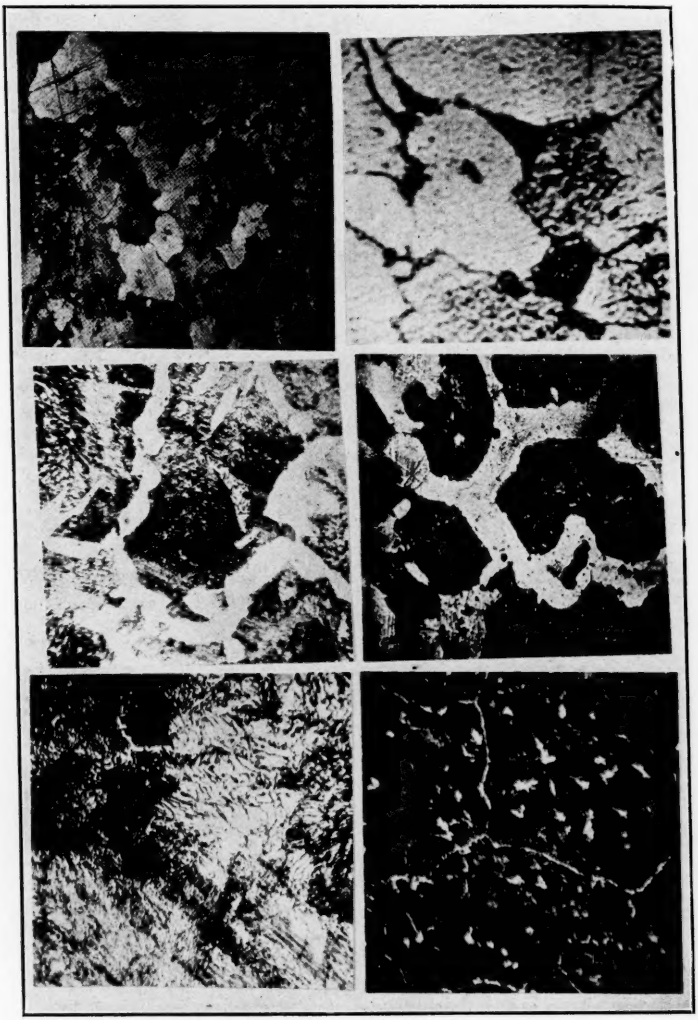


FIG. 70.—Ingot iron.
FIG. 71.—0.10 carbon steel.
FIG. 72.—0.36 carbon steel.
FIG. 73.—0.50 carbon steel.
FIG. 74.—0.85 carbon steel.
FIG. 75.—1.50 carbon steel.

Plate X.

resiliency, lowers the ductility and toughness, distorts the grain and renders it fibrous. The effect of all work may be removed by annealing. Figs. 68, 69, Plate XI.

CARBON STEELS.

Steel may be classified as "Carbon steel" and "Alloy steel." Carbon steel is essentially an alloy of carbon and iron with iron. The structure of the carbon steels will be considered in the order of their carbon content. Further, it should be remembered that steel, unlike iron, is obtained from the molten condition, and is therefore free from slag in layers, and should be free from slag in lumps.

The microstructure of the very low-carbon steels which may contain less than 0.10 per cent. of carbon, as in certain of the so-called "Ingot irons," closely approximates that of a pure metal and appears to be made up almost entirely of ferrite, Fig. 70, Plate X. As the carbon increases, filaments and small patches of pearlite may be noted at the junction of many ferrite grains, as seen in the micrograph of a 0.10 per cent. carbon steel, Fig. 71. With 0.36 per cent. of carbon there is present a greater amount of pearlite, say one-third of the total area, Fig. 72. With 0.50 per cent. carbon about one-half of the area is pearlite, Fig. 73, and it will be noted that the ferrite grains have lost their polygonal form and approximate a net-work.

As the carbon content increases so does the amount of pearlite. It follows, then, that a point of carburization exists at which the mass will be made up entirely of pearlite. This structure is reached with 0.85 per cent. of carbon, Fig. 74. Such a metal is said to be saturated.

A saturated iron carbon alloy is called a "Eutectic steel." One whose carbon content is less than 0.85 per cent. and which contains free ferrite is said to be unsaturated and is called "Hypo-eutectic steel." With more than 0.85 per cent. of carbon the steel is said to be supersaturated and is called "Hyper-eutectic."

Supersaturated steel (and in order to remain a steel it can-

not, according to Howe, contain more than 2 per cent. of carbon), while made up in bulk of pearlite now contains cementite in excess of the amount required to form pearlite. This is seen either in small areas surrounded by pearlite or as a network bounding pearlite areas, Fig. 75, Plate X.

With steels of this character the structure may be more or less confused as a result of heat treatment, and it is often advisable to lightly re-polish after etching with nitric acid and then to re-etch in a boiling 25 per cent. solution of caustic soda in water to which 5 per cent. of picric acid has been added. This colors the cementite black without coloring the other constituents. It may be colored red by a prolonged soaking in a cold 2 per cent. solution of ammonium oxalate.

Cementation and case hardening depend upon the fact that iron heated in contact with carbon absorbs that element. The amount so absorbed depends upon the time and the temperature. Its effect upon the microstructure is greatly to increase the amount of cementite in the carbon-enriched areas. This is the more pronounced at the surface and decreases toward the center. The degree of cementation or case hardening is measured by the amount of cementite present and the depth to which it extends.

In the carbon steels, carbon is the essential constituent, and the other elements, manganese, silicon, phosphorus, etc., may be considered as impurities which may or may not be deleterious. Increasing the carbon increases the tensile strength and decreases the ductility. There is, however, no regular law of increase or decrease, nor is there any proportional relation between strength and ductility. Further, both of these values depend, not only upon the amount of carbon present, but also upon the treatment through which the metal has passed. Thus a quickly-cooled steel has a greater strength and is harder than a slowly-cooled steel.

The influence of carbon on the physical properties is shown diagrammatically in Fig. 77. In this the solid lines represent tensile strength and the dotted lines ductility. Further, the lines marked "A" represent the values for annealed steels, *i.e.*,

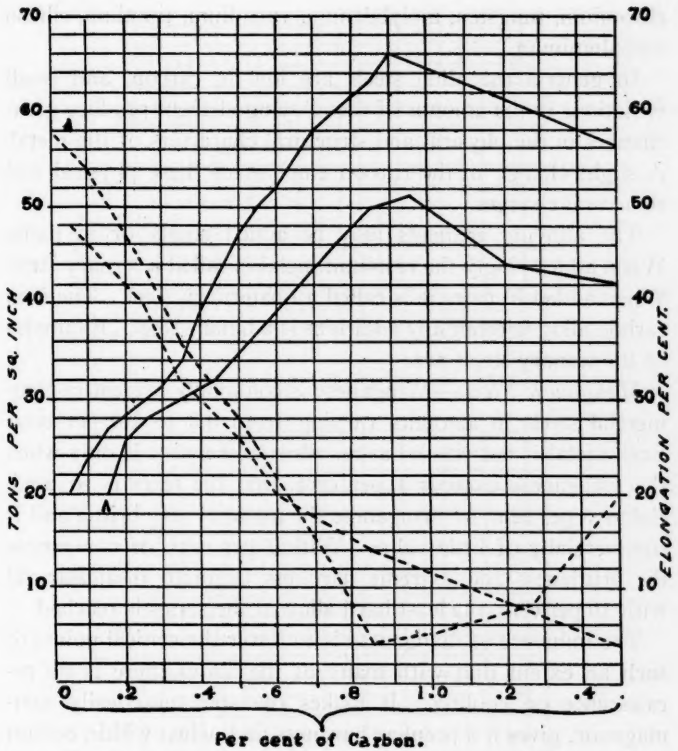


Fig. 77.—INFLUENCE OF CARBON ON PHYSICAL PROPERTIES.

The solid lines represent the tensile strength, the dotted lines ductility. "A" is the value for annealed steels.

those which have been re-heated to the A_1 point and allowed to cool very slowly, while other lines give the values for steels which have not been so re-heated. No values have been plotted for quenched steels, but in general their tensile strength is greater and their ductility is less than that of the untreated and annealed steels.

ALLOY STEELS.

Alloy steels are carbon iron alloys which contain other elements intentionally added to produce certain results. Among the elements thus used are: Manganese, nickel, cobalt, copper,

chromium, tungsten, molybdenum, vanadium, titanium, silicon and aluminum.

In general the alloy steels are low in carbon, and small variations in the amount of the alloying element produce great changes in the physical and structural characters of the metal. A slight change in the carbon emphasizes these physical and structural changes.

The alloying elements may be added singly or in pairs. When added singly the resultant metal is called a ternary steel. When added in pairs it is called a quaternary steel. The iron carbon alloy without any additions is a binary steel. Examples of the ternary steels are:

Manganese Steel.—Manganese is commonly present in commercial steels in amounts varying from 0.3 to 1.0 per cent. Such metal is not classed as an alloy steel and it is only when the manganese exceeds 1 per cent. that the term is so used. Below 5 per cent. of manganese the metal is very brittle and is commercially of little value. With 7 per cent. of manganese the brittleness and extreme hardness begin to disappear till with 10 per cent. the least hard alloy of the series is reached.

The influence of manganese is to lower the critical points to such an extent that with many of the alloys there is no recalescence on cooling. It makes the steel practically non-magnetic, gives it a peculiar hardness and, when within certain limits, increases the strength and elongation.

Heating a manganese steel to 1,000 C. and quenching it in very cold water softens it. Re-heating to 1,000 C. followed by a slow cooling in air restores the hardness. This process with the higher manganese steels greatly increases the strength and ductility. The structure of the low-carbon manganese steels with less than 5 per cent. of manganese is pearlitic. With 5 to 12 per cent. they have a martensitic structure, above 12 per cent. they are crystalline. Increasing the carbon produces the same structure with a smaller amount of manganese; thus with 0.80 carbon a martensitic structure is produced with 3 per cent. of manganese. The relations existing between the

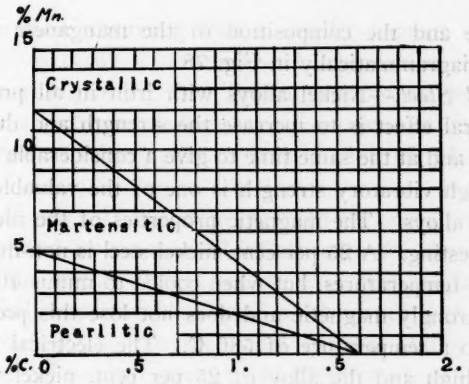


Fig. 78 Structure of Manganese Steels.

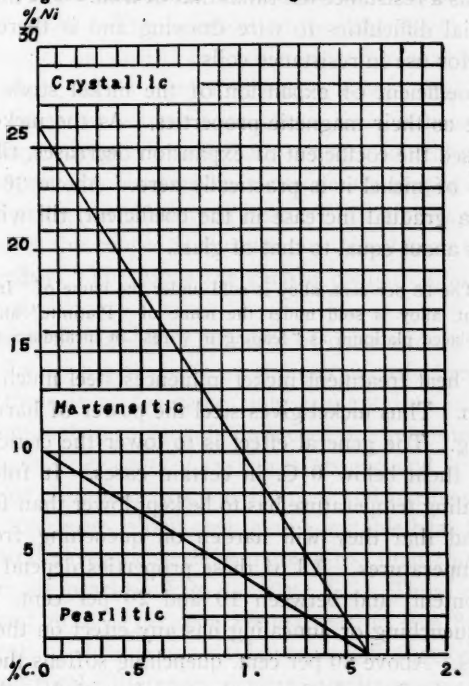


Fig. 79 Structure of Nickel Steels.

structure and the composition of the manganese steels are shown diagrammatically in Fig. 78.

Nickel Steel.—Nickel alloys with iron in all proportions. Its general effect is to increase the strength and ductility of the steel and at the same time to give a considerable toughness Their high vibratory strength is one of the valuable features of these alloys. The magnetic properties of the nickel steels are interesting. A 25 per cent. nickel steel is non-magnetic at ordinary temperatures, but when cooled to minus 40 C. it becomes strongly magnetic and does not lose this property till heated to a temperature of 580 C. The electrical resistance is very high and the alloy of 25 per cent. nickel with 0.80 carbon has a resistance ten times that of iron. The metal offers no especial difficulties to wire drawing and is therefore well adapted for use in resistance coils.

The coefficient of expansion of the nickel steels is as remarkable as their magnetic properties. As the nickel content is increased the coefficient of expansion decreases, till with 36 per cent. of nickel it is practically zero. Above 36 per cent. there is a gradual increase in the coefficient, till with 46 per cent. it is about equal to that of glass.

NOTE.—The 36 per cent. alloy is sold under the name of "Invar." The 46 per cent. alloy is sold under the name of "Platinite" and has very largely replaced platinum as "leading in wires" in incandescent lamps.

Under heat treatment nickel influences steel much the same as carbon. Thus nickel gives steel the power of hardening on quenching. The general effect is to lower the critical points, bringing them below 0 C. in certain cases. It follows that the annealing temperature has to be kept lower than for carbon steels, and that they will harden on quenching from much lower temperatures. All of these properties depend upon the nickel content, and between 10 and 20 per cent. of nickel neither quenching or annealing has any effect on the physical properties. Above 20 per cent. quenching softens the alloy.

The structure of the low-carbon steels with less than 5 per cent. of nickel is pearlitic. With 10 to 25 per cent. it is martensitic. Above 25 it is crystalline. Increasing the carbon pro-

duces the same structure with a smaller amount of nickel. The relations between structure and composition are shown diagrammatically in Fig. 79. Typical pearlitic, martensitic and crystalline structures in alloys having the same carbon content but different nickel contents are seen in Figs. 80, 81, 82, Plate XII.

Cobalt Steels.—It would be expected that cobalt would act like nickel and have a marked influence on the physical properties of the steel. This is not the case, and the increase in the physical properties is so small as compared with the ordinary carbon steels that the metal has no commercial use. Structurally the cobalt steels, up to 25 per cent. of cobalt, are pearlitic.

Copper Steels.—The information available as to the influence of copper on steel is contradictory. Personal experience tends to show that in very mild steels 2 per cent. of copper increases the tensile strength and elastic limit without affecting the ductility. Copper lowers the two upper critical points and in this respect acts like carbon, consequently copper steels will not stand any great amount of heat. With high-carbon steels a portion of the copper is thrown out of solution to form a separate alloy. Therefore the carbon content should always be kept low.

Chromium Steels.—The influence of chromium in general is to raise the tensile strength as the amount of chromium increases; there is at the same time a slight loss in ductility. With a low-carbon content the influence of chromium is small. When about 1 per cent. of carbon is present and from 2 to 4 per cent. of chromium, a great stiffness and hardness without a loss of toughness is obtained upon a suitable heat treatment.

Structurally chrome steels containing less than 7 per cent. of chromium are pearlitic, Fig. 83, Plate XII. Those containing above 7 and up to 15 per cent. are martensitic, Fig. 84. Above 15 per cent. of chromium a double carbide is formed, Fig. 85. These structural differences depend simultaneously upon the percentage of chromium and carbon. Thus in an alloy containing 0.80 carbon the structure is no longer pearlitic with but

3 per cent. of chromium. These structural relations are shown diagrammatically in Fig. 86, Plate XIII.

Heat treatment converts the pearlitic steels into the martensitic kind on quenching and increases their hardness. The double carbide and martensitic varieties are more or less softened.

Tungsten and molybdenum steels furnish pearlitic and double carbide structures. Martensite can be obtained in the pearlitic steels by tempering, and in the double carbide steels by heating to 1,000 C. and air cooling. Both tungsten and molybdenum steels are self-hardening. Further, the effect of molybdenum is more pronounced than that of tungsten, and 1 per cent. of molybdenum is equal to about 4.5 per cent. of tungsten. The relation between composition and structure of these steels is shown diagrammatically in Figs. 87 and 88, Plate XIII.

Vanadium Steels.—Vanadium, like tungsten and molybdenum, gives pearlitic and double carbide steels, which may be grouped structurally under three heads.

The pearlitic series, Fig. 89, Plate XI, which with a low-carbon content, contain not more than 0.7 per cent. of vanadium and with a high-carbon not more than 0.5 per cent. They show an increase of tensile strength, yield point, elongation and reduction of area over the carbon steels with the same carbon content. Heat treating and quenching increases the hardness in proportion to the vanadium content.

The pearlitic and double carbide series, Fig. 90, Plate XI, which with a low-carbon content contain not less than 0.7 nor more than 3.0 per cent. of vanadium, and with high-carbon from 0.5 to 7.0 per cent. In this group the physical properties decrease in value as the vanadium increases. Heat treatment followed by quenching has but little effect, and in general the alloys are inferior to the pearlitic series.

The double carbide series, Fig. 91, Plate XI, which with a low-carbon content contains over 3.0 and with a high-carbon over 7.0 per cent. of vanadium, have very low values for their physical properties, and heat treating is without effect.

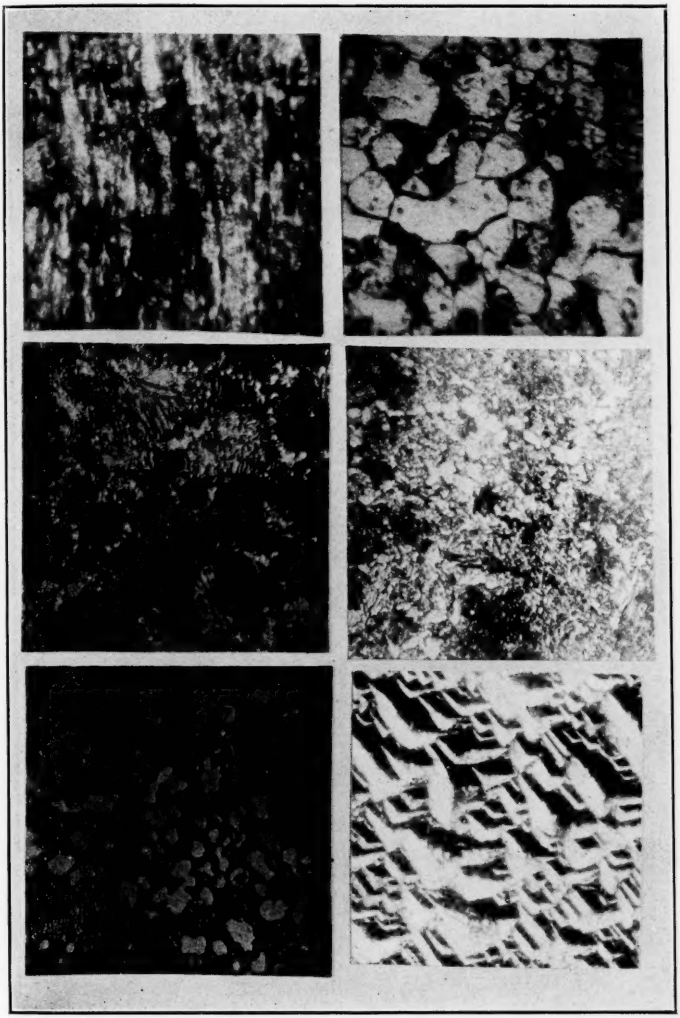


FIG. 68.—Distortion of grain due to a low finishing heat.
FIG. 69.—Grain restored by annealing.
FIG. 89.—Pearlitic vanadium steel.
FIG. 90.—Pearlitic and double-carbidic vanadium steel.
FIG. 91.—Double carbidic vanadium steel.
FIG. 92.—Silicon steel (Stead).

Plate XI.

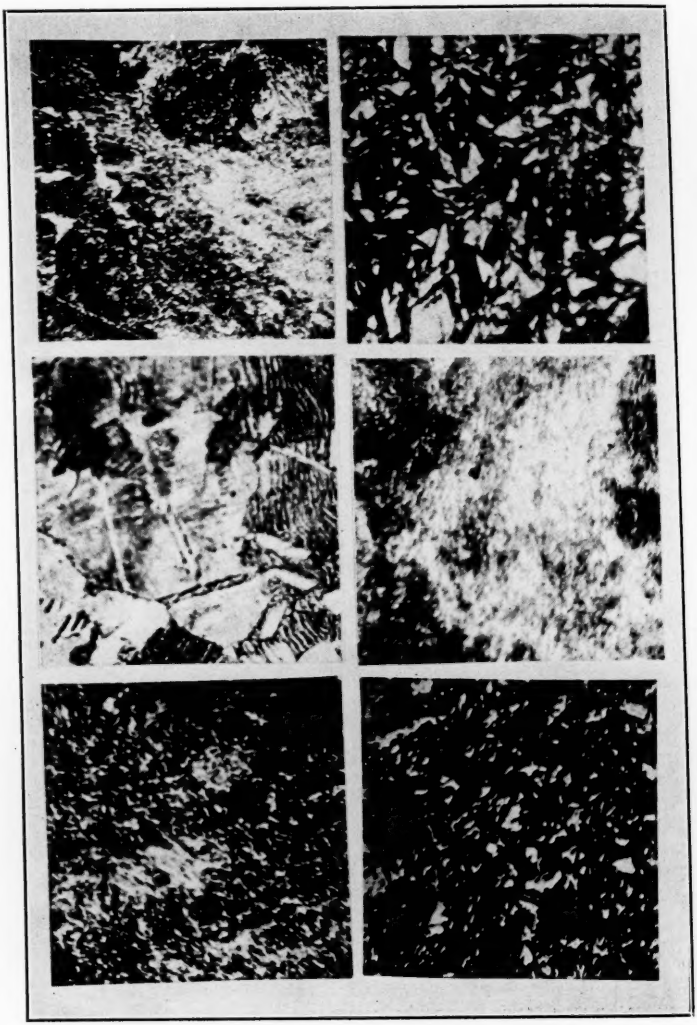


FIG. 80.—Pearlitic nickel steel.
FIG. 81.—Martensitic nickel steel.
FIG. 82.—Crystallic nickel steel.
FIG. 83.—Pearlitic chrome steel.
FIG. 84.—Martensitic chrome steel.
FIG. 85.—Double carbidic chrome steel.

Plate XII.

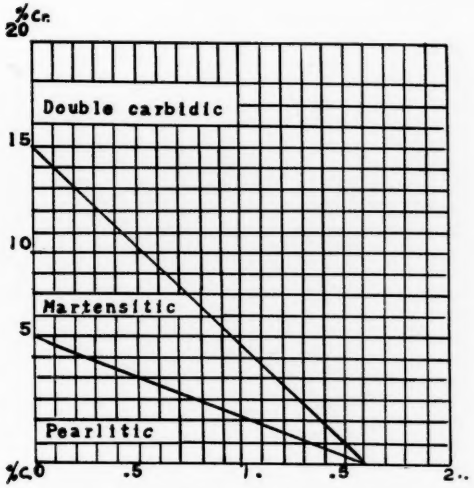


Fig.86 Structure of Chrome Steels.

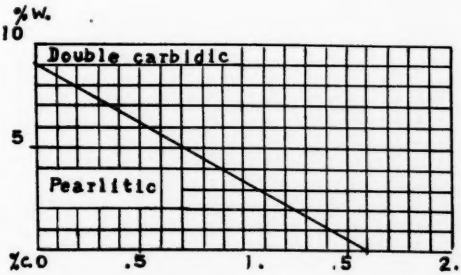


Fig.87 Structure of Tungsten Steels.

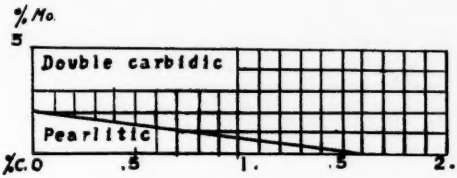


Fig.88 Structure of Molybdenum Steels.

Plate XIII.

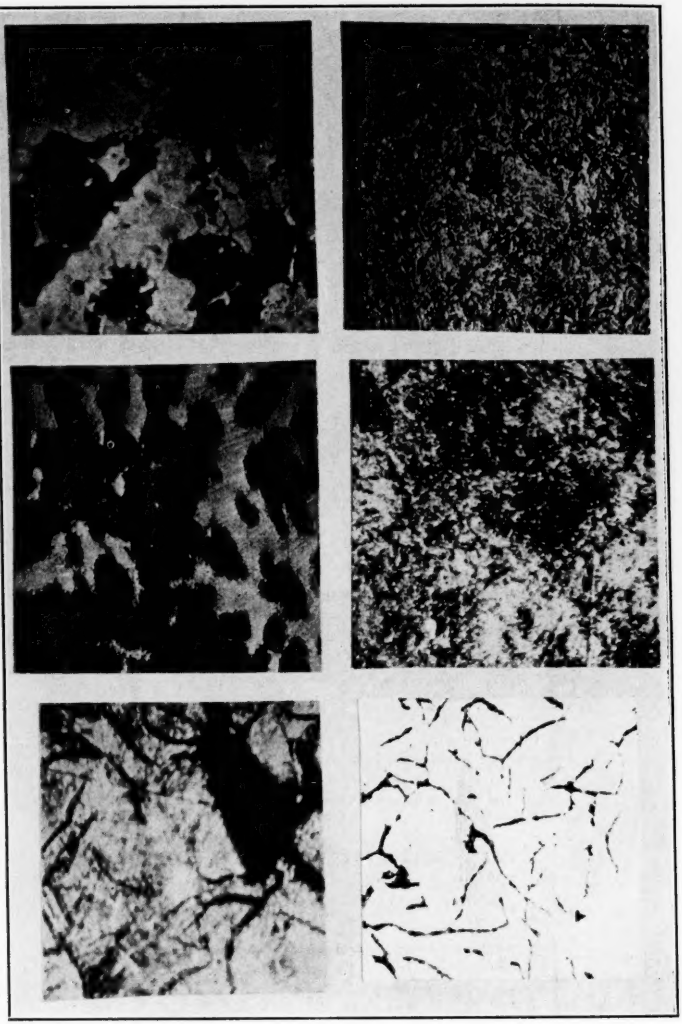


FIG. 93.—Aluminum steel showing graphitization.
FIG. 95.—Chrome tungsten steel as air hardened.
FIG. 99A.—Graphite in cast iron.
FIG. 94. Nickel chrome steel.
FIG. 96.—Chrome tungsten steel as oil hardened.
FIG. 99B.—Graphite in cast iron.

Plate XIV.

The first group is practically the only one of importance, and it should be borne in mind that vanadium steels are materially weakened by work, and it should always be followed by an annealing with slow cooling in order to restore their full physical values. Prolonged heating has a tendency to graphitize vanadium steels, and this should always be looked for, as such steels are treacherous.

Titanium Steels.—These are not to be confused with steel which has had a certain amount of ferro titanium added to the molten metal as a de-oxidizer and de-gasifier. Steels alloyed with titanium up to 10 per cent. have been made. The improvement in their physical properties is so small as compared to the carbon steels that they have not come into use.

Silicon Steels.—Silicon, up to 1.50 per cent, increases the tensile strength and elastic limit without loss of ductility. Above this per cent. the ductility and resistance to shock steadily diminish.

The structure of silicon steels is independent of the amount of carbon. Silicon combines directly with iron to form a silicide which enters into a solid solution with ferrite. It does not form carbides with iron. Above 5 per cent. silicon graphitizes the carbon in the steel, and with 7 per cent. the whole of the carbon will be found as graphite. Such steels are valueless. Silicon steels with 4 per cent. silicon have a large grain and a large crystal size. Fig. 92, Plate XI.

Aluminum Steels.—These are not to be confounded with steel that has had a certain amount of aluminum added to the molten metal as a quieter, de-oxidizer or de-gasifier. Aluminum increases the size of the grain, and with 6 per cent. the alloy is very coarsely crystalline. It acts like silicon, though to a less extent, and, like silicon, causes a graphitization of the carbon, Fig. 93, Plate XIV.

Examples of the quarternary steels are—

Nickel-chrome Steels.—The structural relations of these steels are the same as for the nickel steels as modified by chromium. These influences either act simultaneously, or one is greater than the other according to the proportion of the

alloying elements. It follows that pearlitic, martensitic, double carbidic and crystallic steels, or steels that are at the same time both crystallic and double carbidic, may be obtained. In the case of nickel-chrome steels very little has been done to determine the relation of structure to physical properties. The field is complex and yet may be very fruitful, especially with reference to the efficiency of armor plate, projectiles, high-speed gears, etc. The structure of a steel containing Ni, 2.00; Cr, 1.50; C, 0.40; is shown in Fig. 94, Plate XIV. Heat-treating data is lacking, yet from the structure it is seen that the nickel is the toughening agent and the chrome-carbon the hardening agent.

High-Speed Steels are quarternary alloys of tungsten or molybdenum with chromium, sometimes vanadium, and carbon. Practically nothing has been done to co-relate the physical properties of these steels with their structure and composition. The structure of a chrome-tungsten-vanadium steel as air hardened is shown in Fig. 95, Plate XIV, and as quenched in hot oil in Fig. 96. A comparison of these structures shows at once that the latter is the more efficient as a cutting tool. The machine tests gave the following results, which are the average of 8 trials:

	Feed.	Depth of cut.	Cutting speed.	Length of cut in inches.	
Oil quenched...	1/32	3/16	80	12	
	1/32	1/16	180	12	
Air quenched..	1/32	3/16	80	7.5	failed
	1/32	1/16	180	9	failed

NOTE.—In this series of tests the lathe was stopped when a 12-inch cut had been made.

In general the effect of the added element on the iron-carbon alloy is—

Manganese and nickel lower the critical points and give pearlitic, martensitic and crystallic structures.

Chromium lowers the critical points and gives pearlitic, martensitic and double carbidic structures.

Molybdenum, tungsten and vanadium give pearlitic and double carbidic structures.

Aluminum and silicon are soluble in the ferrite, they have a tendency to graphitize the carbon and increase the grain size.

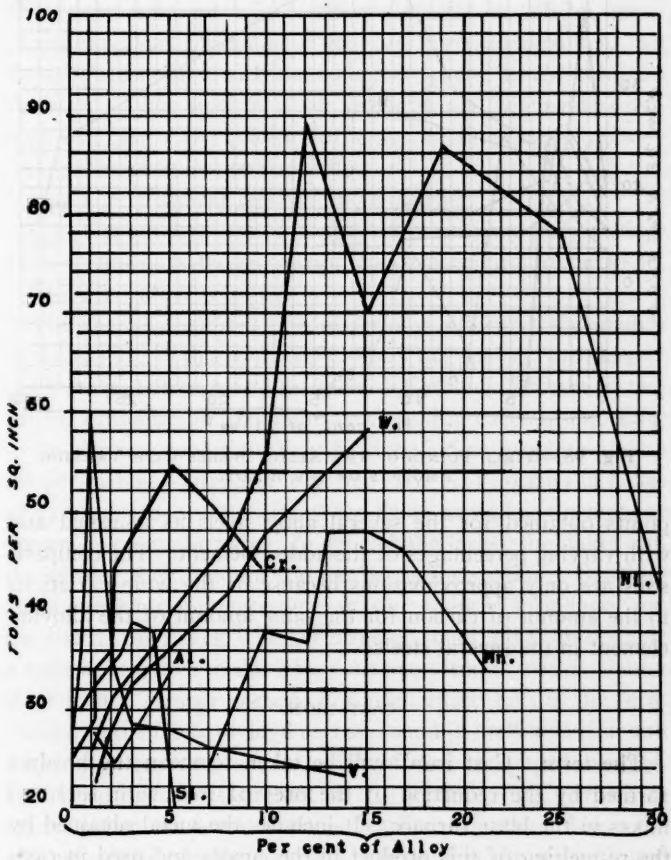


Fig. 97.—STRENGTH OF ALLOY STEELS WITH VARYING AMOUNTS OF THE ALLOY.

The effect of the added element on the physical properties of the iron-carbon alloy is seen in Figs. 97 and 98, which are diagrammatic comparisons of the tensile strengths and yield

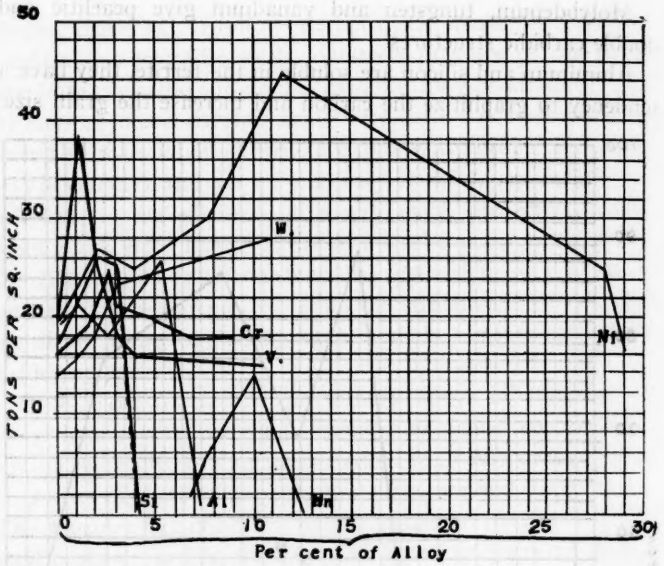


Fig. 98.—YIELD POINTS OF THE ALLOY STEELS WITH VARYING AMOUNTS OF THE ALLOY.

points obtained for the several alloy steels as annealed and with varying percentages of the added element. The comparisons are only approximations because of the wide variations in the amount of carbon for the same amount of the alloying element in the several steels.

CAST IRON.

The term "Cast iron" will be taken to mean the product formed by the reduction of the ores of iron with fuel and fluxes in the blast furnace. It includes the metal obtained by the re-melting of this product in the cupola and used in castings and also the product of the malleablizing furnace. The metal in its composition and structure resembles steel in many particulars and there is no definite boundary between the two. Howe limits the term "cast iron" to that product which contains more than 2 per cent. of carbon. This together with the

foregoing definition will serve to differentiate cast iron from steel.

Cast iron is an alloy of iron with a carbide of iron and carbon. It contains 2 or more per cent. of total carbon with 8 per cent. more or less of other elements, as silicon, manganese, sulphur, phosphorus, etc. Its physical qualities, like those of steel, vary with the composition and heat treatment. Cast iron, unlike steel, is never wrought, and when fabricated is always cast. The qualities of such a casting depend upon—

A.—The chemical composition.

B.—The treatment received in re-melting.

C.—The temperature of casting and the method of pouring.

D.—The character, perfection and temperature of the mold.

E.—The rate of cooling.

The chemical composition materially influences the quality of the casting in that it directly affects the fluidity, shrinkage,* hardness and strength of the metal. Carbon is present in two forms, graphitic and combined. Graphitic carbon, according to its physical condition, may be classified as graphite and as temper carbon. Graphite, Fig. 99, Plate XIV, occurs in flakes of varying sizes and the size of the flakes as well as the amount present is of importance. In general the larger the flakes and the greater their amount the weaker is the casting. Graphite has a softening influence on cast iron, reduces shrinkage, and the size of the flakes, whether large or small, gives an open or a close appearance to the iron. An open iron is weaker than a close iron. Temper carbon, Fig. 100, Plate XV, is the term applied to the finely divided carbon found in malleable iron and which has separated as a result of the prolonged heating. Combined carbon is present as the carbide Fe_3C, which is found either as cementite or as pearlite, Fig. 101. Upon the amount of the combined carbon is dependent the strength and hardness of the metal. It lowers the melting point, increases the shrinkage and gives a close grain to the iron. A soft iron

*Shrinkage is not to be confused with contraction. The one means the pulling or drawing away of the liquid or plastic portion of the metal as a result of the contraction of some portion which has already solidified. The other means the decrease in size as a result of cooling.

may contain from 0.01 to 0.45 per cent. of combined carbon, the harder irons may contain 0.45 per cent. and upwards, and in chilled irons all of the carbon in the chill is in the combined state.

Silicon may be present in cast iron in any amount from a trace to many per cent. For foundry irons in general the amount of silicon required to produce a hard iron is not greatly in excess of 1.50 per cent., for a medium iron not much over 1.75 per cent., and for soft irons it will approximate 2.50 per cent. The influence of silicon is two-fold, in that it softens the iron by throwing the carbon out of solution, thereby increasing the amount of graphite and decreasing the amount of combined carbon, but when all of the carbon is in the graphitic state it may harden the iron. Silicon controls the ratio between the amount of graphitic and of combined carbon and therefore controls the strength of the iron. It also regulates shrinkage in proportion to the amount of carbon present and in a similar manner controls fluidity.

Sulphur in small amounts increases the strength and hardness of foundry irons. The maximum sulphur content of a good casting should not much exceed 0.08 per cent.; more than this weakens the metal. Sulphur increases the amount of combined carbon and thus neutralizes the action of silicon.

Phosphorus tends to make an open iron, and hence any amount is an element of weakness. It makes the iron more fluid and thus aids in the production of accurate castings; hence the use of phosphorus iron for art castings. With too much phosphorus it has a tendency to form globules on the surface of the casting, Fig. 102, Plate XV. The phosphide of iron is much less soluble in the ferrite of cast iron than it is in steel. The amount that is soluble is somewhere near 1.6 per cent., and beyond this the free phosphide separates to form a eutectic with the iron, Fig. 103, and with the carbide, Fig. 104, Plate XV. A casting to have strength should not have much over 0.70 per cent. of phosphorus.

Manganese increases the carbon-combining power of the iron and thus tends to make it hard and brittle when present

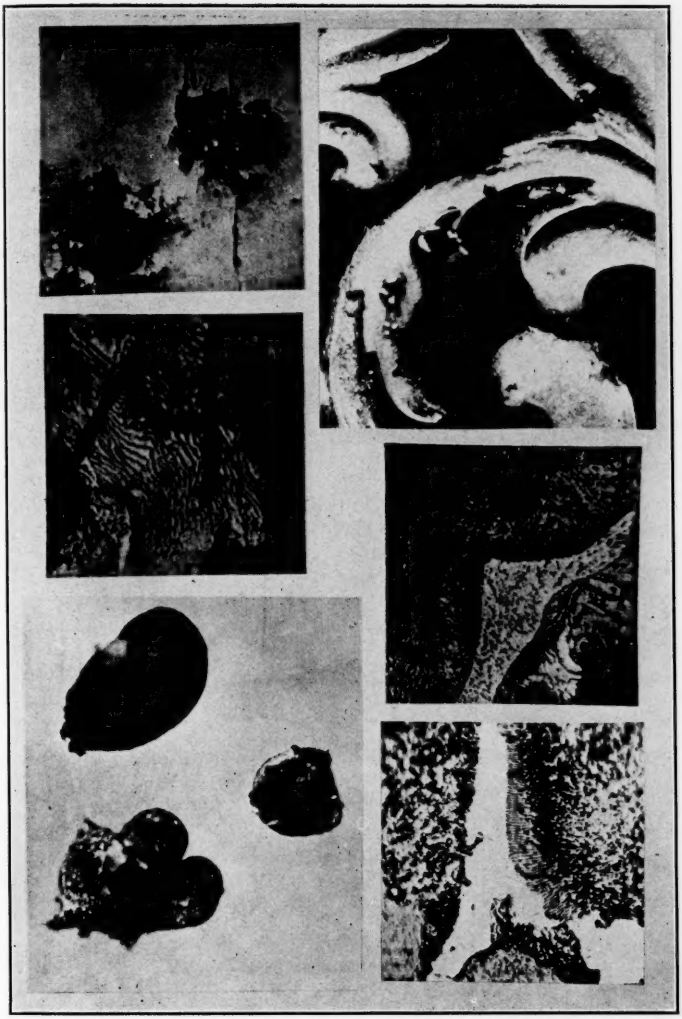


FIG. 100.—Temper carbon.
FIG. 101.—Pearlite and graphite in cast iron.
FIG. 102B.—Phosphide globules out of cast iron.
FIG. 102A.—Phosphide globules on cast iron.
FIG. 103.—Phosphide eutectic.
FIG. 104.—Phospho-carbide eutectic.

Plate XV.

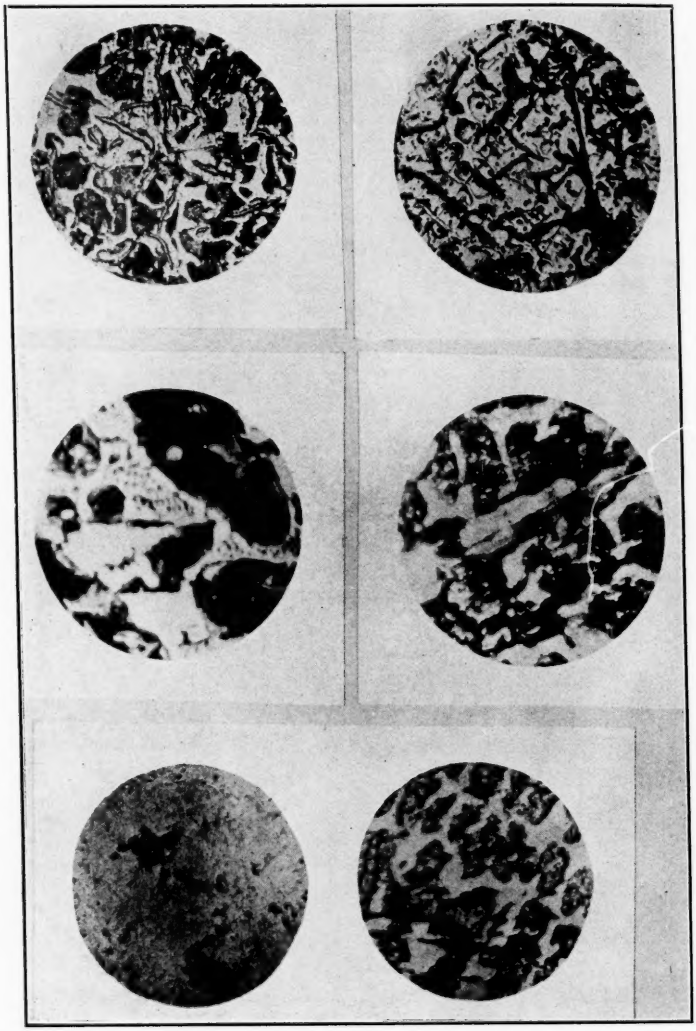


FIG. 109A —Structure of gray cast iron. FIG. 109B.—Structure of gray cast iron.
FIG. 110A.—Structure of white cast iron. FIG. 110B.—Structure of white cast iron.
FIG. 112.—Structure of cast iron after and before malleablizing.

Plate XVI.

in much of an excess over 0.80 per cent. It has a direct action on sulphur and unites with it to form a sulphide.

The elements which give strength to cast iron are: combined carbon, up to a certain amount; manganese, in so far as it affects the sulphur; silicon, in certain cases, through its action on carbon; sulphur, if added to a very weak iron.

The elements which give softness to cast iron are: graphitic carbon; silicon, in that it forms graphitic carbon; and phosphorus, in that it increases fluidity, thus allowing more time for the separation of the graphite.

The elements which give fluidity and low shrinkage to cast iron are carbon, silicon and phosphorus. Fluidity and softness are the most desirable properties in castings not requiring much strength, and iron suitable for such castings should be high in silicon and phosphorus. For castings where a fair strength and a low shrinkage, combined with fluidity and softness, are desired, the determining factors are carbon and manganese to give strength, and phosphorus and silicon to reduce shrinkage. For high-strength castings where shrinkage and softness are factors the amount of manganese should be relatively high and the carbon, phosphorus and silicon relatively low.

The treatment of the iron during melting in the cupola materially influences the properties of the casting. For example, taking the same iron mixture, but in the one case melting more iron to the pound of fuel and in the other less, the first may give hard castings, the other soft castings. The casting temperature and method of pouring both affect the quality of the casting. Hot metal promotes density and hence strength. Proper pouring prevents cold shuts, seams, blows and cavities. The character and perfection of the mold determines the finish of the casting, influences the density, freedom from blow holes, and affects the rate of cooling. The rate of cooling influences strength, homogeneity, density and softness of the casting. It fixes to a degree the condition of the carbon. Rapid cooling tends to give hard white irons, slow cooling soft gray irons. A difference in the rate of cooling of two separate parts of a

casting gives rise to a difference in density and sets up internal strains, which may be remedied by annealing.

The grading of cast iron for foundry use is closely connected with its composition. In the more common practice the grading is based upon the appearance of the fracture of the broken pig, which to a large extent is determined by the condition and the amount of the carbon. The various grades may be broadly grouped in accordance with the appearance of the fracture, as gray, mottled and white. Gray iron contains both graphitic and combined carbon in varying amounts. This variation gives rise to further differences in the appearance of the fracture, which permits of a further classification, as Nos. 1, 2, etc.

Number 1 pig iron has the largest amount of graphite and the smallest amount of combined carbon. It is the darkest in color and has a coarsely crystalline fracture. The iron has a low melting point and is soft and weak. Number 2 iron contains more combined carbon and less graphitic carbon. Its fracture is lighter in color and the grain is less open and more regular, and the iron is harder and stronger, but less fluid than number 1. Number 3 pig iron has a corresponding increase in the amount of the combined carbon. Its fracture is lighter colored, the crystals smaller, and the surface smoother and more compact than number 2 iron. There is a corresponding increase in the strength, hardness and toughness and a decrease in the fluidity.

Mottled iron breaks with a white crystalline fracture through the surface of which is scattered differently sized particles of gray iron, thus giving a mottled appearance. White iron breaks with a white crystalline fracture showing no flakes of graphite. Here all of the carbon is in the combined state.

Structurally the cast irons may be arranged in two groups: One, made up of those which contain graphite, ferrite and pearlite as essential constituents and which are free or practically free from cementite. The other, made up of those irons which consist essentially of pearlite and cementite and which

are free or practically free from graphite. The first group includes the gray irons, the second the white irons.

The microstructure of the gray irons, Fig. 109, Plate XVI, is made up of varying amounts of graphite, ferrite (either silicon or phosphorus ferrite, or both), pearlite, manganese and iron sulphide, and the iron and phospho-carbide eutectics. Number 1 Foundry, which has the least amount of combined carbon, has a correspondingly small amount of pearlite areas and the largest amount of graphite plates. With the increase in the amount of the combined carbon, as in numbers 2, 3, etc., there is a corresponding increase in the pearlite areas, and the highest member of the series is made up essentially of pearlite and graphite. The microstructure of the white irons consists entirely of pearlite and cementite with, of course, some sulphide and phosphide, Fig. 110, and is structurally analogous with a high-carbon steel containing more than 0.90 carbon.

Malleable cast iron is the name applied to that product obtained by the prolonged heating of cast iron at or above 900 C. White iron castings are the more commonly subjected to this process, and its effect is to cause a marked change both in the physical properties and the structure. Its effect on the structure is to convert the cementite and pearlite of the outer portions of the casting almost entirely into ferrite, thus reversing the process of cementation. The transformation of the pearlite and cementite is accompanied by the disappearance of any flake graphite that may have been present in the affected areas and by the appearance of temper carbon, as the finely divided graphitic carbon is called.

EFFECT OF THE USE OF JACKETS, THROTTLING AND THE VARYING OF STEAM PRESSURES AND EXPANSIONS ON THE ECONOMY OF MARINE ENGINES.

By F. W. Milner, Lieutenant, U. S. N., Member.

Experiments have been made to determine the effect of varying pressures, of cut-offs and of jackets on the performance of steam engines by Isherwood, Willans, Thurston, Gately and Kletzsch, Emery and Loring, and many others.

Most of these experiments were performed to find out the general law of the effect of each element. The general results were uniform enough to indicate that—

1. The earlier the cut-off the greater the economy.

2. Condensation is a function of the number of revolutions, and increases as the power is reduced.

3. Jackets are of use in some cases and in others they are a source of loss.

4. Throttling is generally a source of loss.

5. With a decrease of power, the water rate of an engine increases.

6. The cut-offs or number of expansions have an important effect on the economical development of power.

Still, with the effect of all these individual factors fairly well understood, they merely indicate the way towards the economical construction and operation of steam engines, and it is difficult for the operating engineer to determine precisely under what conditions for a given power his engine will perform most efficiently.

Although a large number of experiments have been made on the stationary steam engine, the number of exhaustive trials of marine engines is few. The test of a marine engine is much more difficult than of a stationary engine, due to lack

of room and to the limitations imposed by the service of the vessel.

The results obtained from tests of the former do not afford much data that can be used for marine engines. On stationary engines the revolutions can be varied independently from the indicated horsepower, while with a marine engine, for a certain number of revolutions, the I.H.P. is a non-variable quantity, other conditions being equal. For instance, a dynamo generally runs at constant speed and the I.H.P. may vary from almost zero to a maximum. On the other hand, a marine engine running at a certain number of revolutions develops a certain power, and the revolutions cannot be changed without changing the power developed and, *vice versa*, the power cannot be changed without changing the revolutions. Furthermore the type of engine is quite different.

Perhaps the most complete and accurate experiments performed on stationary engines are those of Gately and Kletzsch which are described in the "Journal of the Franklin Institute," Vol. 90. These were for the purpose of determining the individual effects of pressure, piston speed and expansion upon condensation. These experiments indicated—

1. That condensation increases as the number of expansions is increased, the pressure remaining constant, and the horsepower varying.

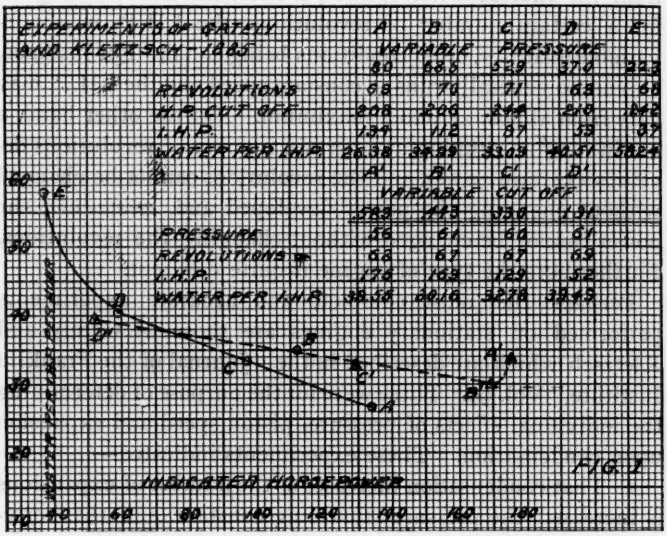
2. Condensation increases with decrease of boiler pressure, revolutions and cut-offs being constant but I.H.P. varying.

In Fig. 1 are shown the results of these tests. It can be seen that it is hard to apply these to a marine engine. The only condition from which a comparison might be drawn is where the revolutions and the indicated horsepower are nearly the same in both cases, as is shown in the following table:

	Pressure, pounds.	Cut-off.	Revs.	I.H.P.	Water rate.
"A"	80	.208	69	134	26.38
"C"	60	.330	67	129	32.78

It is seen here that an earlier cut-off and a higher pressure give a better result than a lower pressure and a late cut-off, which does not agree with most of the experiments performed

to ascertain the value of different cut-offs. This is about the only comparison that can be made with these curves, for nothing is known of what the results would have been had the I.H.P. and the revolutions been kept in a certain ratio, as they are in a marine engine.


EXPERIMENTS OF CATLIN AND ALEXANDER - 1885
A B C D E
VARIABLE PRESSURE
80 68.5 53.2 37.6 22.3
REVOLUTIONS 68 76 71 68 68
H.P. CUT OFF .808 .206 .144 .108 .082
I.H.P. 137 112 87 53 37
WATER PER I.H.P. 26.38 28.89 33.05 40.51 58.24
A' B' C' D'
VARIABLE CUT OFF
283 213 136 131
PRESSURE 65 61 60 61
REVOLUTIONS 68 67 67 69
I.H.P. 174 168 129 52
WATER PER I.H.P. 28.58 28.16 32.78 38.45
INDICATED HORSE POWER
FIG. 1

The main point to be noted is that *it is possible at a certain power to obtain different water rates by varying the pressure, cut-offs, etc.*

From the experiments made on the Revenue Cutters *Bache*, *Rush*, *Gallatin* and *Dexter*, Emery adopts an empirical formula for the best ratio of expansion, which may be written $r = 1 + \frac{p}{22}$, where p is the absolute pressure in pounds per square inch, and r the number of expansions.

Emery states that the results from this formula are "nearly correct for single engines of large size, with details of good design; too large for engines of ordinary construction, and too small for the better class of compound engines."

Some writers have attempted to apply this to triple-expansion engines. Such an empirical formula cannot apply to all types of engines, for the number of economical expansions is affected by piston speed, type of engine, etc.

ISHERWOOD'S EXPERIMENTS ON THE "SIESTA."

In 1883 Chief Engineer Isherwood, U. S. Navy, performed a series of experiments on the Herreschoff yacht *Siesta* to determine the economical effect produced—

1. By closing the cut-off valve of the L.P. cylinder at different points of the stroke of its piston.
2. With different measures of expansion.
3. With different initial pressures on the piston of the H.P. cylinder.
4. By simultaneously increasing the speed of the pistons and the pressures upon them.

The *Siesta* was a composite vessel 100 feet long, 15 feet 2 inches beam, 6 feet draught and of 63 tons displacement. She had a vertical, unjacketed, two-cylinder compound engine, 10½ inches by 18 inches by 18 inches stroke. She had three-ported flat slide valves, operated by Stephenson link, with a cut-off valve working on the outside of the flat valve. This allowed variations in cut-off, but did not affect the admission, release or compression. Steam was supplied by a coil boiler.

These experiments were made with the usual accuracy of Isherwood.

Accurate measurements of feed water, coal and ash were made. Indicator cards were taken at intervals of ten minutes; tests were of three hours' duration, except one, *J*. I have selected only the important data which are shown in Table I.

RESULT OF EXPERIMENTS.

In comparing the different results the comparisons should be confined to cases where the piston speed, initial pressure and back pressure do not vary materially.

1. *The Effect of Changing the Cut-off in the L.P. Cylinder.*

TABLE I.—TRIALS OF "SIESTA" BY CHIEF ENGINEER ISHERWOOD, U. S. NAVY, TO FIND ECONOMIC EFFECT OF VARIOUS CUT-OFFS, MEASURES OF EXPANSION AND STEAM PRESSURE. THROTTLE WIDE OPEN, EXCEPT IN EXPERIMENTS F, G, H AND I, WHEN IT WAS PARTLY CLOSED.

	A	B	C	D	E	F	G	H	I	J	K	L
Revolutions.....	146.5	169.4	170.3	190.2	177.9	178.5	173.3	177.1	176.6	193.1	185.3	184.4
Abs. initial pressure in H.P. at beginning of stroke.....	110.3	102.9	105.3	105.2	96.5	82.2	80.4	78.3	91.3	66.7	68.6	64.9
Abs. pres. at cut-off...	94.86	90.71	90.13	83.33	75.55	74.02	74.9	72.34	76.3	61.95	59.22	58.85
Cut-off in H.P. cyl...	.153	.226	.236	.241	.417	.400	.410	.436	.448	.297	.871	.870
L.P. cyl...	.263	.251	.465	.208	.237	.452	.846	.857	.874	.298	.871	.870
Back pressure, abs....	3.79	3.76	3.34	5.122	4.33	4.36	4.07	3.78	5.09	5.77	5.64	5.26
Expansions..... ...	12.9	9.9	9.6	5.9	6.2	6.4	6.28	5.97	5.84	3.24	3.26	3.26
I.H.P., H.P. cyl.....	32.0	38.0	48.1	46.5	37.8	40.6	59.9	62.0	69.8	27.9	54.9	56.3
I.H.P., L.P. cyl.....	34.7	46.6	40.1	84.7	63.1	55.6	33.0	34.8	37.0	113.0	58.2	62.8
Total I.H.P.....	66.7	84.9	88.1	131.2	100.9	105.3	92.9	96.8	106.9	141.0	113.1	119.2
Feed water per I.H.P.	17.92	18.87	17.91	16.78	18.53	18.02	20.61	20.59	19.22	20.04	24.83	23.81

—From trials B and C we find that the initial pressure, pressure at cut-off, expansions, back pressure and piston speed are practically the same, the only difference between B and C being in the position of the L.P. cut-off, which in B is at .251 of stroke, and in C at .465. The water rate of B is 17.91 pounds, and of C is 18.53 pounds per I.H.P.

In F and G the conditions are approximately the same, except that in F, with a L.P. cut-off of .452, the water rate is 18.02 pounds, and in G, with a cut-off at .846, 20.61 pounds. The later cut-off gives a greater water rate in each case. Comparing E and I, which give greatest range of cut-offs, we find that with a L.P. cut-off of .237 the water rate is 18.53 pounds, and with a cut-off of .857, 19.22 pounds.

From these experiments it is seen that *the position of the L.P. cut-off has a marked effect upon the economy of operation of an engine, and that the later the cut-off the greater the water rate per I.H.P.*

2. *The Influence of Number of Expansions on the Economy of an Engine.*—In B and E we have, respectively, 9.9 and 6.2 expansions. In the former the water rate was 18.87 pounds, and in the latter 18.53 pounds. The greater expansion ratio gave a greater water consumption.

In the other experiments differences in the conditions prevent comparisons. Isherwood deduced from these experiments that "no economic gain could be obtained by expanding the steam more than four to five times, even in a compound engine using saturated steam of a very high pressure, in unjacketed cylinders, with pistons working at a very high reciprocating speed."

This deduction is no longer quite correct, for it has been proved that steam can be economically expanded a greater number of times than six, even in a compound engine.

The point to be observed is this: that with a certain engine running at a certain piston speed, and having a certain initial pressure, *there is a certain number of expansions which produce a maximum economy, and that a number of expansions greater or less than this, will increase the water rate.*

3. *The Effect of Greater Initial Pressures.*—In experiments H and I all conditions are the same except initial and back pressure, and the results are:

	Initial pressure, pounds absolute.	Back pressure, pounds absolute.	Water per hour per I.H.P.
H, ...	78.3	3.78	20.59
I, ...	91.3	5.09	19.22

GENERAL RESULTS OF ALL EXPERIMENTS.

All these experiments show that the efficiency of an engine varies considerably with the conditions of its use. The efficiencies of different engines vary with type of engine, dimensions of cylinder, thickness of metal, back pressure, initial pressure, piston speed, expansion of steam, jacketing, valve gear, etc., so that deductions from one set of conditions with a given engine may not apply to other conditions for the same engine, and still less with other engines. Each case is a separate problem which requires a distinct solution.

EXPERIMENTS IN THE BRITISH NAVY.

In 1899 accurate experiments were conducted on the British cruisers *Argonaut*, *Diadem*, *Arrogant*, *Hermes* and *High Flyer*. The water-tube boiler had just been installed on men-of-war, and a great increase of pressure was made available. The question of the value of high pressure, the proper ratio of cylinders, the value of the jacket was then unsettled, and these experiments were accordingly undertaken.

The first experiments were carried out on the contractor's trials of the *Argonaut*, viz: thirty hours at 3,600 I.H.P., thirty hours at 13,500 I.H.P., and the full-power trial of eight hours at 18,000 I.H.P. The *Argonaut* is a twin-screw cruiser of 18,000 I.H.P., with four-cylinder, triple-expansion, vertical engines 34 inches × 55.5 inches × two 64 inches × 48 inches stroke, with cylinder ratios of 1 :: 2.66 :: 7.09. Steam pressure at engines was 250 pounds, and r.p.m. at full power 120.

The barrels only were jacketed, and these drained to the main condensers. H.P. and I.P. valves were of piston type, and the L.P. valves were double-ported slide valves. Stephenson link motion was used. The reversing gear was of the type used in our engines. Each cylinder had an independent linking-up adjustment.

Steam consumption of the main engines was determined under various conditions, as regards jacketing of cylinders, steam pressure, and rates of expansion. Observations were also made on the steam consumption of auxiliaries, but these were not the principal objects of the tests, so they were not very extensive.

To measure the water from the main and auxiliary condensers the hot-well pumps were made to discharge to a pair of calibrated measuring tanks fitted on deck. These were alternately filled and emptied. It was arranged that the only water which passed to the main condenser was that from the engines, jackets and separator. The water from the separator was measured, so that the economy of the various methods of working the main engine could be exactly ascertained. The water from auxiliaries was first measured in separate tanks, and then pumped to the tanks on deck.

The valves in full gear, as measured, gave the following cut-offs in their respective cylinders: H.P., 73 per cent.; I.P., 67 per cent.; L.P., 46 per cent.

The results of the trials are given in Tables II, III and IV. For convenience the trials are arranged in order of number of expansions.

Referring to Table II, the trials at full power, the expansions were the same in each case. The only variable is the use of the jackets. The pressure carried in the jackets was about 20 pounds above receiver pressure, except in the H.P., where it was kept at the same pressure as in the receiver.

The results were as follows:

C. All jackets in use, water per I.H.P. per hour, 16.64 pounds.
B. L.P. only, . . water per I.H.P. per hour, 16.15 pounds.
A. No jackets, . . water per I.H.P. per hour, 15.75 pounds.

In other words, the use of the jacket increased the consumption .9 pound, or 5.3 per cent., at full power. As it is easily possible to make an error in computing the I.H.P., the water per revolution per hour is a better indication of the relative economy, except in cases where there is a marked variation in the number of revolutions, as in trial G, of Table III. The data given under this heading are comparable only when the I.H.P. or revolutions are approximately the same.

	Revolutions.	Water per I.H.P.	Water per revolution.	Tons water per day.	Coal per day.*
C.	129.2	16.64	2,453	3,395	377.2
B.	129.2	16.15	2,364	3,265	362.8
A.	127.7	15.75	2,315	3,165	351.7

The difference between A and C shows that the use of jackets makes a difference of 25 tons coal used per day, an amount worth considering.

Table III, at 13,500 H.P.—Expansions varied from 12.5 to 8.3. In trials G, C and D jackets were used in various combinations. In the other trials no jackets were used. For distributing power in G, C and D both the main and H.P. independent links, in F only the main links, in B only the H.P. independent links, were used. In A and E the links were in full gear. In B there was considerable throttling, otherwise the results would probably have been better. This is shown by a comparison of A and E, which are identical in all conditions except pressures. In A the pressure was reduced from 194 pounds at the the boiler to 170 pounds at the engine, and in E from 232 pounds at the boiler to 181 pounds at the engine. The results show that the lower boiler pressure is by far the more economical; that throttling in this case resulted in a serious loss. The general results may be shown better as follows:

	G.	C.	D.	F.	B.	A.	E.
Pressure in boiler.....	280	280	251	275	287	194	232
engines.....	246	235	230	235	189	169	180
Expansions.....	12.5	10.4	10.4	9.7	9.1	8.3	8.3
Water per I.H.P.....	15.85	15.87	15.44	15.36	15.97	15.73	16.22
day, tons.....	2,140	2,424	2,277	2,387	2,314	2,308	2,398
Coal per day, tons*.....	238	269	253	265	257	256	266

* Allowing 9 tons water per ton of coal.

TABLE II.—FULL POWER.

Position of valve gear to get power.	Jackets.	Revs.	Pressure.		Vacuum.	Cut-off cylinder.			Expansion.	I.H.P.	Water per hour per I.H.P.	Per revolution.	Total water per hour.
			Boiler.	Eng.		H.P.	I.P.	L.P.					
C. Main links full, H.P. run in	All	129.2	299	238	26.0	71	67	46	8.5	19,046	16.64	2,453	316,990
B. Main links full, H.P. run in	L.P	129.2	290	241	25.8	71	67	46	8.5	18,946	16.15	2,364	305,527
A. Main links full, H.P. run in	Nil	127.7	292	236	26.0	71	67	46	8.5	18,781	15.75	2,315	295,718

TABLE III.—THIRTY HOURS AT 13,500 I.H.P.

G. Main and H.P. Ind. links	All	112.4	280	246	26.6	43	48	31	12.5	12,631	15.85	1,790	200,154
C. Main and H.P. Ind. links	I.P., L.P.	117.0	280	235	26.6	54	57	38	10.4	14,279	15.87	1,936	226,548
D. Main and H.P. Ind. links	Nil	115.3	251	230	26.6	54	57	38	10.4	13,778	15.44	1,846	212,873
F. Main links only	Nil	116.9	275	235	27.0	60	52	35	9.7	14,521	15.36	1,916	223,027
B. Main full H.P. right in	Nil	116.7	287	187	26.5	65	67	46	9.1	13,556	15.97	1,855	216,422
A. All links full	Nil	116.6	194	168	26.2	73	67	46	8.3	13,718	15.73	1,850	215,747
E. All links full	Nil	116.4	232	179	26.5	73	67	46	8.3	13,813	16.22	1,926	224,092

TABLE IV.—THIRTY HOURS AT 3,600 I.H.P.

A. Main and H.P. links	All	75.9	285	166	26.0	28	34	23	15.7	3,783	16.32	814	61,749
B. Main and H.P. links	I.P., L.P.	75.5	294	167	25.9	28	34	23	15.7	3,843	16.34	831	62,798
C. Main and H.P. links	Nil	75.0	275	169	25.7	28	34	23	15.7	3,762	16.26	814	61,195
D. Main link only	Nil	75.6	281	142	25.7	39	34	23	13.2	3,779	16.68	835	63,120
E. Main full, H.P. run in	Nil	77.1	285	78	26.2	65	67	46	9.1	3,847	17.58	877	67,531
F. All links full.	Nil	76.4	130	68	26.4	73	67	46	8.3	3,867	17.72	897	68,514

H. M. S. *Argonaut.*

Although G seems to give the best economy, it cannot be admitted in the comparison, as the revolutions were 112.3 as against the average of 116.2.

The use of jackets at this power shows a loss of about .4 pound per I.H.P., or 2.6 per cent. at 13,500 H.P. In Table IV, at 3,600 H.P., where the expansions varied from 15.7 to 8.3, the use of jackets caused but about one-half of one per cent. loss. Here the jackets begin to be of use, and at a still lower power would probably result in considerabe gain.

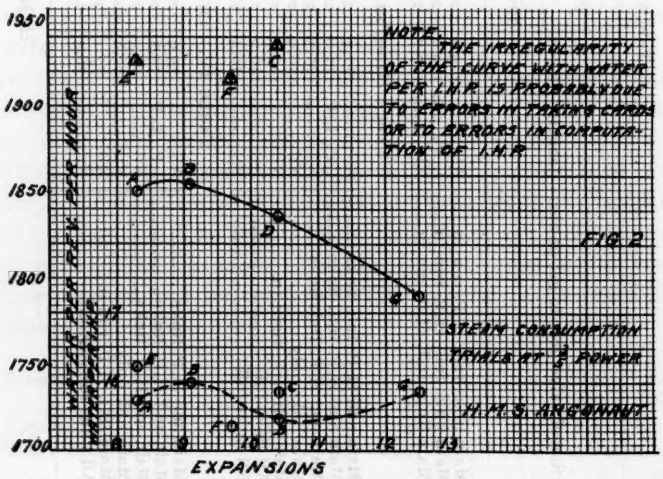
The general deductions from this set of experiments are:

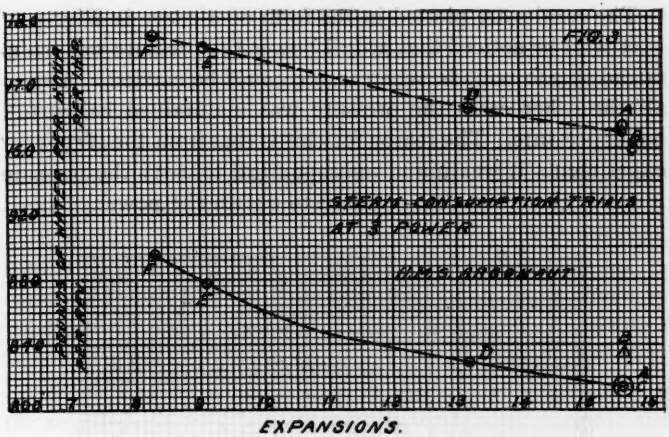
1. The jackets are a source of loss above one-fourth of the maximum power.

2. That in this case the economy varies directly as the number of expansions is increased.

3. That at a given number of revolutions and a given number of expansions, the lower the initial pressure the greater the economy.

The results of Tables III and IV are shown in Figs. 2 and 3.





EXPERIMENTS ON H. M. S. "HERMES."

For data down to one-tenth power, corresponding to cruising speed, trials were carried out on H. M. S. *Hermes*, a second-class cruiser of 5,600 tons displacement, 10,000 I.H.P., with two triple-expansion, 4-cylinder engines, with cylinders 26 inches × 42 inches × two 48 inches × 30 inches stroke, or in the ratio of 1 :: 2.61 :: 6.81. The ratio $\frac{6.81}{1}$ is very low, and corresponds to that generally found in merchant ships. This engine was designed for a low water rate at the lower powers. These trials were a continuation of those of the *Argonaut*, and the same means of getting accurate data were used as in the trials of that vessel. A summary of the results is shown in the following table:

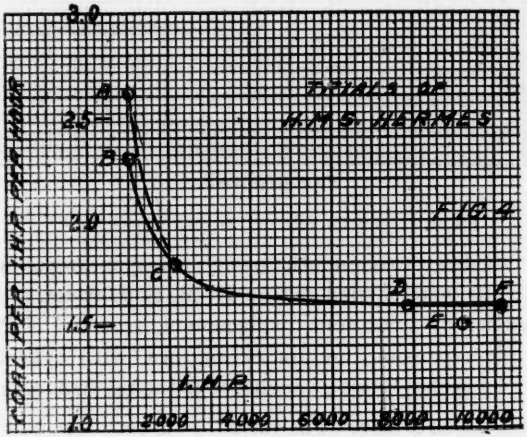
H. M. S. "HERMES."

	Twelve hrs. at 1,000 I.H.P.	Twelve hrs. at 1,000 I.H.P.	Thirty hrs. at 2,000 I.H.P.	Thirty hrs. at 7,500 I.H.P.	Eight hrs. at 9,000 I.H.P.*	10,000 I.H.P.
Pressure (boiler)	239	166	196	264	270	240
(engine).....	172	124	126	223	247	229
Cut-off H.P., per ct..	20	50	26.2	56	56	71
Vacuum.....	26.1	26.8	26.5	25.5	25.1	24.5
Revolutions	84.9	86.5	110.9	165.7	175.8	182.2
I.H.P.....	1,074	1,018	2,151	7,713	9,203	10,224
Coal per I.H.P.....	2.61	2.31	1.34	1.59	1.52	1.58

* Auxiliary exhaust to L.P. receiver.

The results are plotted in the form of a curve in Fig. 4.

It is to be noticed that at full power the pressure was maintained at 240 pounds, whereas 300 could be carried. Also in all the other cases the pressure was maintained below full boiler pressure. The reason for this is that from the *Argonaut* experiments it was found uneconomical to carry the pressure much higher than that required to obtain the power.



Comparing the two twelve-hour trials at one-tenth power, which are most important, we find that the revolutions are practically the same, the vacuum the same, the differences being in cut-off and pressure carried. In the first trial the pressure was throttled from 239 to 172 pounds, and the cut-off in the H.P. was at two-tenths stroke, giving 34 expansions. In the second case, the pressure was throttled from 136 to 124 pounds and the cut-off in the H.P. cylinder was at half stroke, giving about 13.6 expansions. Contrary to the general theory that the earlier the cut-off the greater the economy, the reverse proved to be true. The pressure in the second trial was greatly reduced, and this contributed to the greater economy. The consumption includes that for auxiliaries and the lower pressure reduced their consumption. Thus, steam-

ing at the same speed under one set of conditions gives a total coal consumption per day of 30 tons, and the other a consumption of 25.2 tons, a difference of 4.8 tons per day, or a difference of 16 per cent.

It is also shown that in this case at 1,000 I.H.P. the most economical number of expansions is about 13.6, and an increase in number will cause a serious loss.

In another set of trials of the *Argonaut*, in the same year, similar experiments were carried out to find the most economical cut-off and pressures for both main and auxiliary machinery. There were six of these trials, each of five hours duration. During the first two trials the auxiliaries were operated from the same boilers as the main engine, while in the remaining four trials separate boilers were used for the auxiliaries. The summary of the data is as follows:

Period.	Steam pressure.	Cut-off in H.P. cyl., per cent.	I.H.P.	Coal for main engine.	Coal for auxiliaries per I.H.P. main engine.	Total coal per I.H.P.
1st	155.5	51.3	2,096	2.34	0.45	2.79
2d	213.5	23.8	2,147	2.3	0.58	2.88
3d	225	23.8	2,086	2.02	1.06	3.08
4th	234	33.9	2,229	2.09	0.82	2.91
5th	160	51.3	2,240	2.03	0.78	2.81
6th	161.5	60.0	2,312	2.12	0.80	2.92

From this it can be seen that the most favorable individual and over-all results were obtained with a pressure of about 160 pounds and a cut-off of about 51 per cent. This is another example of the most economical cut-off not being the earliest attainable.

STEAM CONSUMPTION OF AUXILIARIES, H. M. S. "DIADEM."

On the H. M. S. *Diadem*, during the same year, a series of eight-hour trials was made to determine the steam consumption of auxiliaries; by a series of eliminations the water consumption of each auxiliary being determined. After this was done two tests of the auxiliaries, which consisted of—

1 blower,
1 circulating engine,
1 dynamo,
2 distiller pumps,
2 evaporators,

were made with boiler pressures 280 and 160 pounds per gage, other conditions being the same. The coal consumption per day was—

Boiler pressure, 280 pounds; coal per day, 10.4 tons.
160 pounds; coal per day, 8.8 tons.

or a difference of 15 per cent.= 1.6 tons.

The results obtained on all these trials agree with those trials of our own scout cruisers, which indicated that the steam consumption of an auxiliary, using steam non-expansively, varies directly as the revolutions plus a fraction of the density of the steam. In other words, the greater the pressure the greater the steam consumption.

TESTS OF AUXILIARIES, U. S. S. "SALEM."

The following tables of the steam-consumption trials of the flushing pump and a main-feed pump of the U. S. S. *Salem*, from which have been plotted the curves in Figs. 5 and 6, confirm this:

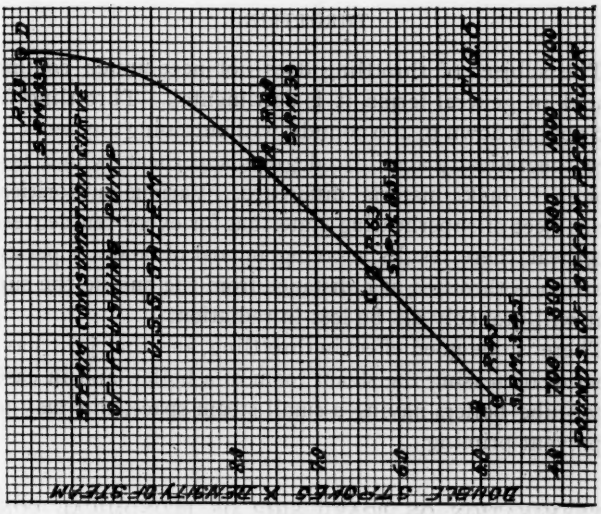
FLUSHING PUMP.

	Boiler. Pounds.	Pressures. Steam chest. Pounds.	Vaccum.	Back pressure.	Disc. pres.	Double strokes.	I.H.P.	Water per hr.
A...	200	88	10	6.3	50	33.0	7.6	959.3
B...	200	45	10	6.0	20	34.5	3.8	573.5
C...	187	63	10	6.7	30	35.3	5.9	827.3
D...	191	73	10	5.9	30	53.3	9.5	1,088.6

MAIN FEED PUMP.

	Boiler. Pounds.	Pressures. Steam chest. Pounds.	Vacuum.	Back pressure.	Disc. pres.	Double strokes.	I.H.P.	Water per hr.
A...	241	165	10	5.9	300	12.8	19.8	1,775.3
B...	241	203	10	6.2	300	24.8	41.7	2,917.0
C...	218	211	10	6.2	300	34.3	52.8	4,185.6
D...	251	240	10	6	300	45.3	72.1	5,446.7
E...	257	223	10	6	300	34.3	52.4	3,934.9

It is preferable, therefore, to use steam at as low a pressure as possible when operating auxiliaries.



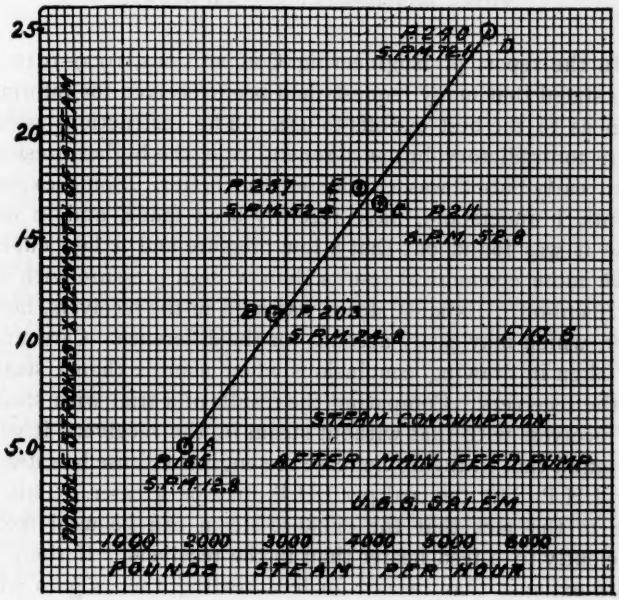

25
20
15
10
5.0
STEAM CONSUMPTION
AFTER MAIN FEED PUMP
U.S.S. SALEM
1000 2000 3000 4000 5000 6000
POUNDS STEAM PER HOUR

FIG. 6

THROTTLING.

The subject of throttling does not need much comment. Stated in general terms, it is detrimental. Throttling from 250 pounds down to 150 pounds, without doing any work, will give theoretically about 10 degrees superheat, which is not enough to make material difference. Various authorities on the subject of superheating state that for every ten degrees of superheat there results a saving in coal of about 1 per cent. If steam is throttled at a distance from the engine the superheat will probably all be lost in radiation.

There is one good effect in throttling, and that is, it has the effect of increasing the steam room of a boiler, and in case of water-tube boilers having a small steam space, this may tend to prevent priming.

DETERMINATION ON BOARD SHIP OF ECONOMICAL CONDITIONS FOR OPERATING MACHINERY.

In the report of Engineering Competitions, 1909–1910, it was stated that many requests had been received for information as to the best cut-offs to use. The instructions given were contradictory, but as a general rule, the higher pressure with early cut-off was considered the better. Some experiments, I understand, have recently been made, and in one case it was found out that a low pressure and a late cut-off gave more economical results than a high pressure with an early cut-off. Exhaust steam was used in the receivers, however, which may easily account for the difference. Care must be taken in drawing conclusions when using exhaust steam in the receivers. Comparing the results of a trial with steam in receivers with that without steam in the receivers, it will be found that in the former case the engine will use less steam per I.H.P., but the coal per I.H.P. may be greater. This is due to the fact that the exhaust steam can be used more efficiently in a feed heater than in the receivers. If there is more exhaust than needed for feed heating, then a gain will

be made by sending the excess exhaust steam into the receivers.

From the experiments which have been cited it can be seen that pressures, cut-offs, use of jackets, and throttling, exert a great influence on the economical development of power in a marine engine. It is therefore very important for the senior engineer officer of a ship to ascertain this data for all speeds, and especially for the cruising speed.

Theorizing will not prove as much as will practical experiments. In our own Navy we have made a departure from an established system by adopting large ratios of expansions. Beginning with the *South Carolina*, a ratio $\frac{\text{L.P.}}{\text{H.P.}}$ of $\frac{10}{1}$ was made and which has since continued. Now the *Argonaut* had a ratio of $\frac{\text{L.P.}}{\text{H.P.}}=\frac{7}{1}$. The *South Carolina* had independent linking-up gear giving the following ranges of cut-offs:

H.P., .44 to .78 of stroke.
I.P., .43 to .70 of stroke.
L.P., .38 to .60 of stroke.

On the *Argonaut*, with a cut-off at .44 in the H.P. cylinder, the nominal number of expansions would be $\frac{7.09}{.44}=16$; on the *South Carolina* 22.8. Both vessels carry about the same boiler pressure, 295 pounds. The question of the most economical number of expansions would seem to be the more important in the case of the *South Carolina*. In the *Argonaut* trials, even by running in the main links, no more than 15.7 expansions could be obtained, and the limit of economical expansion was not reached. In the *Hermes* it was very conclusively shown that the most economical number of expansions at one-tenth power was about $\frac{6.81}{.5}=13.6$, and that an increase up to 34 expansions was exceedingly uneconomical.

The Navy Regulations require that the senior engineer officer endeavor to determine the most economical cut-off of

the engine at cruising speeds. From information given in the Engineering Competions, it is noted that the question is still unsettled, and a general plan of attack of this problem may be interesting.

THE MOST ECONOMICAL OPERATING CONDITIONS.

The question of the most economical operating conditions is extremely important to the engineer who wishes to be able to steam a certain distance on the smallest amount of coal. It does not take a great number of experiments nor a very long time to be able to determine them. By some engineers it is stated that they found no difference in economical results at a certain speed, no matter what were the arrangements of pressure, jackets and cut-offs. Such may be the case with some engines, but it can not be stated as a rule applying to all. As stated by Isherwood, all sorts of theories may be advanced regarding the limits of expansion, etc., but theory can not stand up against the result of carefully conducted experiments.

Quite often, due to drills, etc., it is almost impossible to find time to perform these experiments. If such be the general situation, then it would seem advisable to instruct commanding officers to hold such trials, particularly upon commissioning, and these trials might be covered in the Rules for Engineering Competition.

The performance and coal-consumption curves made from data of standardization and acceptance trials furnish practically no data. The links on the standardization trials are kept at full gear. The boiler pressure is generally high. All regulation of power is made by throttling. The following is a scheme suggesting a way to ascertain these desired data.

1. Calibrate feed tanks.

2. Arrange so that the only steam entering the condenser of the engine to be tested is the steam from the engine and from its jackets.

3. Adjust cut-offs of each cylinder to properly distribute the work.

4. Trials:

(*a*) Four without throttling and without use of jackets.

(*b*) Two with jackets.

(*c*) Two with maximum throttling.

5. Plot results.

6. Determine effect of steam to receiver.

7. Determine effect of auxiliaries.

CALIBRATION OF FEED TANKS.

As each feed tank is to be used as a measuring tank, it should be accurately calibrated. This may be done by pouring in known quantities of water and graduating the gage glasses, or by filling the tank first and drawing off definite amounts. In each case there should be a graduated scale on the side of the tank, making it possible to read the volume of the contents at any time.

Feed tanks are constructed so as to hold enough water to last at least five minutes at full power. At one-tenth power the tanks would hold enough water to last thirty minutes. With the exhaust of one engine and its jackets only running into the feed tank, it would take about an hour to fill it. While a test of an hour may not give accurate results, the comparative results obtained on different trials would be of great value.

ENGINE AND JACKET STEAM TO FEED TANK.

The arrangement for allowing only the engine and jacket steam to go to the calibrated tank of the engine under test is simple. The cross connection between tanks is closed; one feed pump only need be used, that one having suction from the other feed tank. All auxiliary exhaust is put in the other feed tank.

SETTING OF CUT-OFFS.

Isherwood, in his experiments on the *Siesta*, found out that the position of the L.P. cut-off made considerable difference on the economical development of power, but this trouble is spared us because we are required to divide the total power in the proper proportion amongst the four cylinders.

The proper setting of the independent links to distribute the power equally between the three expansions should be determined for each speed. If not at hand this should be found out before taking up the steam-consumption test. A curve should be plotted showing the relative setting of the links at a certain power. For instance, suppose on a certain ship steaming at 65 revolutions per minute the following arrangements of cut-offs are found to divide the work up proportionally:

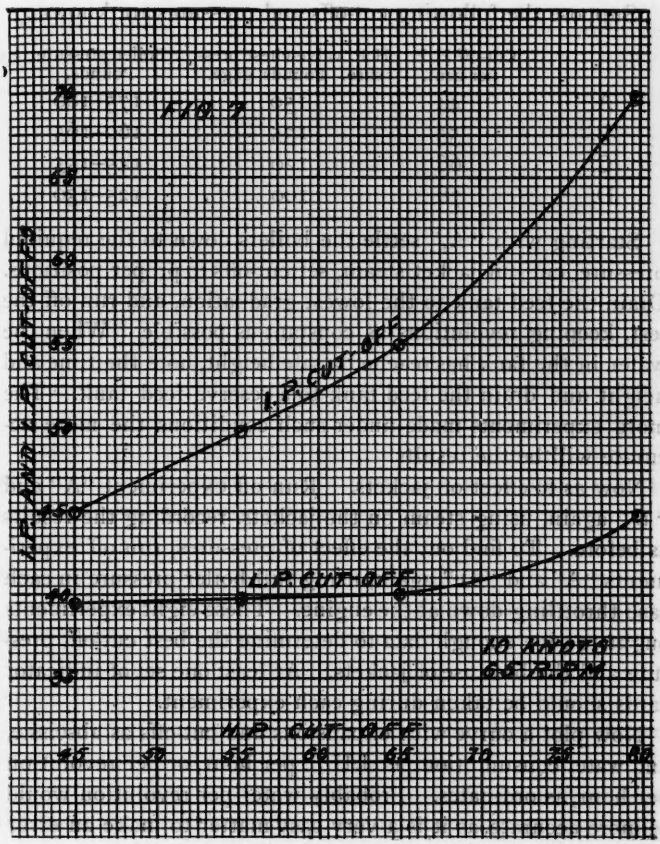
H.P.	I.P.	L.P.	
.78	.70	.44	65 revolutions.
.66	.55	.40	2,000 horsepower.
.55	.50	.38	10 knots.
.45	.45	.38	

Plotting the values of H.P. cut-off as abcissae, and the values of I.P. and L.P. as ordinates, we have the curve on the following page.

TRIALS.

When the previous arrangements have been made the trials can be undertaken. To get absolutely accurate data a great many experiments should be performed, but eight trials will suffice to point out the characteristics of the water-consumption curve.

(*a*) Suppose that we have an engine whose H.P. cut-off can be varied from 44 per cent. to 78 per cent. of the stroke. Take four points, say 44 per cent., 55 per cent., 66 per cent. and 78 per cent., and let these be the only points of cut-off used. Now as to pressures. From previous experience, when arriving at the data concerning the relative setting of the in-



dependent links, the pressure needed in the H.P. valve chest at each setting can be ascertained. These pressures being known, keep about 25 pounds more on the steam line in each case, throttling at the throttle valve to get the required valve-chest pressures. This is necessary in order to insure steady revolutions, and also to allow for increased power for a short time in case of emergency. Sometimes it will be impossible to reduce the pressure at the boilers below a minimum determined by the requirements of the auxiliaries.

Suppose the following cut-offs and pressures are known:

	Cut-off H.P., per cent.	H.P. valve-chest pressure. Pounds, gage.	Boiler pressure. Pounds.
A,	44	150	175–180
B,	55	135	160–165
C,	66	120	145–155
D,	78	100	125–130

Starting with "A," settle the boilers down to the required pressure, cut out jacket steam, set cut-offs to proper positions. Drain the feed tank to the lowest graduation, and run off the test, keeping revolutions as constant as possible. When the water in the tank has reached a safe height, note the time and height and drain the feed tank to a reserve feed tank. The more experiments made under such conditions the more accurate will be the results.

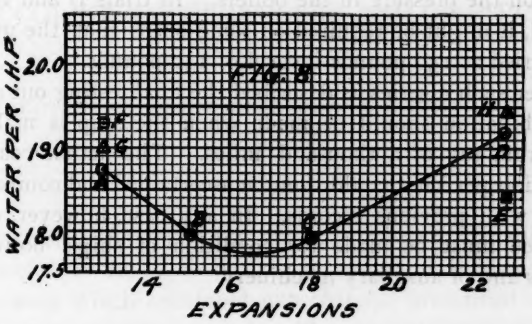
Now as to effect of jackets. Run off experiment "A" and "D" again, using steam on the jackets, recording the results as before. To find out the effect of pressures, run off experiments "A" and "D" again, using full boiler pressure in each case throttling down at the engine. These eight experiments ought to give enough data to determine the best cut-off, pressure and the best use of jackets. At least one set of indicator cards should be taken during each experiment.

Now plot a curve with expansions and cut-offs as abscissæ, and water consumption per I.H.P. per hour, or per revolution per hour, as ordinates. Probably water per revolution would be more accurate, as it is a very easy matter to make an error in taking cards or in computing the horsepower.

Suppose, for example, we get the following result:

	Conditions.	Rev.	Pressures. Boiler.	Eng.	Vac.	H.P. Cut-off.	Water per hour. Total.	Per I.H.P.	Per rev.
A...	No jackets...	75	180	150	25	.44	37,600	18.8	501
B...	No jackets...	75	165	135	25	.55	36,000	18.0	480
C...	No jackets...	75	155	120	25	.66	35,600	17.8	475
D...	No jackets...	75	130	100	25	.78	38,400	19.2	519
E...	Jackets.....	75	130	100	25	.78	36,800	18.4	491
F...	Jackets.....	75	180	150	25	.44	38,600	19.3	515
G...	Throttling...	75	295	150	25	.44	38,000	19.0	507
H...	Throttling...	75	295	100	25	.78	38,800	19.4	517

Plotting these values on an expansion-water-rate curve we have the following:



Assuming that the above data had been found experimentally, and it is perfectly possible that such a variation can be obtained, we deduce the following:

1. That the most economical number of expansions is about 16.8, corresponding to a cut-off in the H.P. of about .60.

2. That at 12.8 expansions the use of jackets causes a loss, while at 22.7 expansions their use results in material gain.

3. That full boiler pressure, causing throttling to get power, is uneconomical.

While performing these experiments no live or exhaust steam should be admitted to the receivers.

Even after determining the most economical conditions for the main engine the assumption that these insure a maximum economy of the entire plant is not necessarily true, because at low power, say one-tenth, the auxiliaries have an important effect upon the coal consumption. They consume about 30 per cent. of the total steam used at one-tenth power. However, it can be assumed from the experiments on the British cruisers and from the trials of the scout cruisers that the lower the pressure the less is the steam consumption of the auxiliaries, and if the economical rate of the engine requires a low pressure, then this will favor an economical operation of the auxiliaries.

The steam pressure required to operate auxiliaries such as dynamos, steering engine, air compressors, etc., sets a lower limit on the pressure in the boilers. In trials D and E mentioned, it might be required on some ships that the pressure be carried at 150 pounds instead of 130 pounds.

After such a series as described the final testing out of such a combination should be made when the ship is making a long run, say of a thousand miles. The actual coal consumption per knot could then be ascertained and comparisons made with previous records. In any case, however, due allowance must be made for condition of ship's bottom, of boilers and of auxiliary machinery.

THE CORROSION OF BOILERS AND OF PIPING ON SHIPBOARD.

By Lieutenant Commander Frank Lyon, U. S. Navy, Member.

The writer's attention was strongly drawn to this subject by experience in the U. S. S. *Oregon* and U. S. S. *New Jersey*, experience which convinced him that the prescribed methods of preventing corrosion were neither certain nor adequate.

From July, 1896, to August, 1899, as an Assistant Engineer, he served in the *Oregon*, and for the greater part of this time was in direct charge of her boilers. She had fire tubular boilers. The feed water was kept hot in practically an open tank and was at all times kept so strongly alkaline as to render it unfit for ordinary purposes. In this vessel corrosion of boilers or of piping was almost unknown.

He served from May, 1906, to May, 1909, as the Senior Engineer Officer of the U. S. S. *New Jersey*. Troubles from corrosion began immediately and continued throughout the three years. In two months from the time the ship was commissioned the copper suction and discharge piping to the circulating pumps of the dynamo condenser and ice machine corroded through in places. As these pipes are in the dynamo rooms, this rapid action was, as usual, attributed to electrolysis from stray currents. The electrical installation had many grounds, especially in wet weather, as most of the porcelain fittings were porous. Within a year the main and auxiliary injection and discharge piping in the engine rooms had corroded through in many places. This could not be attributed to the galvanic action of stray currents with the same apparent certainty as that in the dynamo rooms, especially as the electric installation had, in the meantime, been made reasonably tight. The main

injection and discharge piping was removed, holes were filled and the inside of pipes given a wash with soft solder. No more troubles were experienced with them.

Within one year and a half the flushing system around the pumps and distillers was renewed in many places; the ice machine and dynamo condenser suction and discharge pipes to circulating pumps had corroded through again. An examination of all of the corroded pipes showed the troubles almost invariably to take place in bends and at or near flanges, the places where the copper pipes had been worked most. The corrosion was most noticeable in the piping in which the water is heated more or less between its entrance to and exit from the system.

As no troubles were ever experienced with the discharge from the air pumps (main, auxiliary, or dynamo), or in the fireroom piping, there was a grave doubt cast on the stray current galvanic action theory.

The *New Jersey* had Babcock & Wilcox boilers. The feed tanks were placed under the main condensers with filter tanks at the top of one end. The dynamos were of the enclosed crank-case type using splash feed, with cylinders just above the crank case, and parts of the valve and piston rods entered both the steam and oil spaces. Originally there was no grease extractor between the dynamo air-pump discharge and the filter tank. Most of the lubricating oil used in the dynamos found its way into the feed tanks and boilers. The water in the boilers was kept slightly alkaline or neutral, using soda for the first year and lime after that time, as it was found that lime floated the oil to the surface better than soda. Due to fear of priming in these small drum boilers, the water was never kept as alkaline as that in the *Oregon's* boilers.

Subsequently grease extractors were installed and the dynamo cylinders were raised so that no part of the rods entered both the steam and oil spaces, and the oil thereafter was kept out of the boilers. Up to this time practically no corrosion had taken place in the boilers and no damage to them had been caused by the oil.

In December, 1907, the screws holding together the ends of the composition packing rings in the water ends of three main feed pumps, broke within a period of four days. These sharp ends shaved up the composition liners of the pumps and pumped them into the feed system. Later some of these chips or shavings were found in the boilers.

Boiler corrosion began, and as it was practically coincident with the pump troubles it was attributed to the ground-up liners that had gotten into the boilers through the feed system and to sea water from leaks in the condensers which developed from time to time. Zincs, the supposed palliatives of all corrosion troubles, were added in greater numbers. The zincs corroded rapidly and so did the boilers. Tubes pitted through, hand-hole gasket seats corroded, and cross and side boxes and uprights suffered. In one period of two weeks, twelve changes of boilers for renewing pitted tubes were made.

Most of the corrosion had taken place in the form of pitting, and it was attributed to galvanic action from particles of composition, ground into the sides of the steel tubes by the turbine cleaners.

When replacing handholes a liberal use of graphite on the gaskets had always been allowed.

It was felt that every effort to prevent corrosion and to keep the boilers clean had been made, yet destructive local corrosion was going on and increasing in effect in spite of the zincs, non-acidity of the water, cleaning and other efforts to stop it. The Navy Regulations had been followed, experiments had been made, yet the writer had failed in every particular to stop corrosion on metals with their surfaces in contact with water. He was detached from that duty and left it, knowing that there was something woefully wrong in the methods he had pursued and with the general methods of treating corrosion on shipboard.

With this excellent grounding in a knowledge of the effects of corrosion, the writer has been engaged for the past three years, at the Naval Engineering Experiment Station, in an experimental investigation of the problem of preventing cor-

rosion. Naturally much of the work has followed lines suggested by other investigators. The purpose of this paper is to present some of the results of this investigation, and with them, to explain why, in the *Oregon,* corrosion was prevented, while in the *New Jersey* it was induced by the methods employed.

While present at an operation at about this time, it was noticed that the nurse insisted on having several spoonfuls of soda put into the basin of water, into which she afterwards dropped the doctor's instruments. She said that that was the only way the instruments could be kept from corroding or tarnishing until they could be cleaned and thoroughly dried, and that every good doctor knew that soda solution would absolutely stop corrosion if it was strong enough, while good distilled water was not good for the purpose. That was the writer's first intimation that there was anything with water in it that was less corrosive than good neutral distilled water.

POTENTIOMETER METHOD.

A beginning at tests was made by determining the difference of potential and the current flowing by the potentiometer method between different metals, or two pieces of the same metal in water and in different solutions under varying temperatures, at the same time measuring the rate of loss of weight per square inch of area. It was learned that:

With two pieces of the same plate of metal, one was at times of higher potential than the other, and at other times a reversal had taken place. The curve of E.M.F. between two pieces of Monel metal from the same bar immersed in boiler brackish water, when plotted on a volt-time basis, made almost a complete curve of sines, and in the same period of time the two pieces lost almost exactly the same weight.

With the steel copper-couple, both metals being kept at the same temperature, the E.M.F. generated increased with a rise in temperature to about 160–180 degrees F. and fell gradually after that.

When the steel in its water was kept at room temperature

and the copper in its water was raised, the E.M.F. increased until the temperature of 165–170 degrees F. was reached, when it began to decrease steadily thereafter. When the copper in its water was kept at room temperature and the steel in its water was raised, the E.M.F. generated increased to a temperature of from 155 to 170 degrees F. and decreased thereafter. The rate of loss on the steel was highest at from 160–170 degrees F. The reasons for this rise and fall of E.M.F. were assumed to be as follows: The difference of potential between the copper and steel is the sum of the difference of potential between the copper and the water and the steel and the water. As the maximum difference of potential and the maximum loss of weight of steel occurred at from 160–180 degrees F. it must be that as the temperature rises the potentials of all three change. At first that of steel rises with increased temperature up to 160–180 degrees faster than that of the water solution; above this temperature that of the water solution rises the faster. The potential of copper decreases or remains the same with rise of temperature up to 155–165 degrees, and above that it rises faster than that of the water solution. It has been proved that at some definite temperature copper has the same potential as steel, while at greater temperatures it is higher.

With sodium carbonate in water and the copper-steel couple kept at the same temperature, as the concentration increased, the volt-concentration and the loss-concentration curve reached zero at the same point. This point was afterwards found to be at about the upper-limit concentration (the next one above the highest one in which corrosion is seen) of soda for steel. Assuming that the metals of the couple remained at the same potential, that of the liquid must have been increased until it had risen to about that at any part of the steel. In different concentrations of other chemical solutions containing sodium as the positive radical, similar results were obtained. The upper-limit concentration varies somewhat with the different grades of irons and steels. These variations are very slight in good, homogeneous, evenly-treated metals and are very much more

in impure, segregated, or unevenly treated metals. They are due to the different potentials of the different impurities and of the particles stressed differently in the surface in contact with water.

The upper-limit concentration of sodium carbonate was less with the same steel at high than at lower temperatures, thereby indicating that the potential of the solution increased with rise in temperature faster than did that of the steel. The upper-limit concentrations of sodium carbonate, potassium chromate and potassium dichromate in distilled water solutions, were about the same as found by Heyn and Barnes and by Cushman, using different methods.

With a polished zinc-steel couple in distilled water, the zinc potential was the higher. With the same steel connected to old unscaled zincs removed from the boilers and condensers of the U. S. S. *Olympia,* the steel was of the higher potential; after the zincs were scaled they were at the higher potential or were electropositive to the steel. With a new zinc-steel couple immersed in distilled water and daily readings taken, at first the zinc was at the higher potential. The difference of potential fell gradually, but the steel remained bright; after the fourth day corrosion in places on the steel was seen and continued to spread daily, while the difference of potential continued to decrease until on the fourteenth day the pieces were at the same potential and both were covered with oxide. On the sixteenth day the zinc was electronegative or lower in potential than the steel and remained so until the thirtieth day, when the readings were stopped. Zinc oxide removed from a zinc from the *Olympia's* boilers was electronegative to the steel.

With the new zinc-steel couple in brackish water, results similar to the foregoing were obtained, it taking a day or two longer to bring about each change.

With the zinc-steel couple in alkaline water solutions, the above actions did not occur when the concentration was above the upper limit for steel. The zinc continued to corrode or to dissolve in all concentrations tried. No upper-limit concentration for zinc was reached.

With a steel-carbon rod couple in distilled and in brackish waters the carbon was always electronegative to the steel.

By plotting volt-time curves from potentiometer measurements it was seen that the voltage changed from time to time with the same metals, very slightly with some metals and more so with others. From this it would seem that the difference of potential measured was that between the highest potential on the surface of the electropositive metal and the lowest on the surface of the electronegative one. That as the surfaces were changed by solution of particles from the surface of the electropositive metal, new particles were exposed, perhaps of higher or lower potential.

EXPERIMENTS WITH THE FERROXYL MOUNT.

Investigations of the same nature were taken up, using the Ferroxyl mount of Cushman and Walter. In this method the metals under examination are covered with a clear, neutral jelly while the jelly is hot. The jelly contains two indicators, (1) ferro cyanide of potassium, which combines with the metal at the points where it is dissolving into solution (the electropositive spots), giving a color reaction characteristic of the metal with which it combines; and (2) phenolphthalein, which combines with the hydroxyl (OH) where the current is entering the metal from the liquid (the electronegative spots), giving the characteristic reaction of phenolphthalein in alkaline solutions. The same effect can be produced in a metal immersed in water if the water is not shaken enough to diffuse the colors. The mount in jelly can be preserved almost indefinitely if the surface of the jelly is kept covered with alcohol.

By means of these mounts it may be seen that:

On a good piece of steel or iron the potentials at different points in the surface are continually changing, while on a piece containing slag and segregated impurities or areas over which the molecule stresses are different, the potentials are more or less permanent. On steel this mount gives a blue compound at points where corrosion is going on.

A piece of scale from an old zinc removed from a boiler is electronegative to boiler steel and will cause it to corrode.

A new zinc-steel couple in the water mount shows the zinc adequately to protect the steel for about four days, then gradually to lose its effect until at from the fourteenth to the sixteenth day the protective action is entirely lost. The zinc then corrodes locally, while the steel corrodes over its entire surface.

Zinc scale from old zincs was seen to be electronegative to three grades of boiler steel, three grades of nearly pure irons, to four grades of cast iron; and to Admiralty condenser tubes, Muntz metal, Tobin, Phosphor, and Manganese Bronzes, and to Naval composition. It was electropositive to Monel metal, sheet copper and to Elephant Bronzes.

Zincs that had just been cleaned and polished were at first entirely electropositive to steel boiler plate; after five days' immersion they were electropositive in some spots and negative in others to steel, electronegative to greater areas of the steel after ten days, and to the entire surface of the steel after twenty days.

Steel boiler tubes that had been properly expanded into the tube sheet, that is, evenly over the entire thickness of the tube sheet, were shown to be in about the same condition as that of the unexpanded tube. With tubes in which the metal was stretched in expanding, due to improper placing of the expander rolls, pronounced permanent electropositive spots were produced throughout the thickness of the tube at the places where the metal was deformed over the edges of the tube or header sheet, and also when the surface of the tube was cut by the expander rolls.

Specimens that had been broken in the testing machine, showed themselves to be strongly electropositive in the breaks, and where the grips had squeezed or cut the metal, as compared with parts where the skin had not been broken.

In a machine steel rod that had been broken in a torsion machine the neutral axis could be followed for some distance from the break by its being electronegative to the outer or badly twisted sections.

Some grades of irons and steels were seen to be electro-positive to others, and these results were confirmed by measurements on the potentiometer.

Graphite placed on steel surfaces showed it to be electro-negative to the steel.

The difference of potential and current flowing was roughly measured on a potentiometer between a blue and a red area on the same plate of steel. The method indicated that the points were of different potential, but did not indicate the relative amounts of such differences or any means of accurately measuring them.

ACID CORROSION TEST.

The relative corrosions of irons and steels were tried in distilled water and in acid solutions as follows:

American ingot iron, wrought iron, steel boiler plate and cast-iron specimens were corroded in distilled water and a rate of loss per unit of area per unit of time established for each. They were then removed and placed in the same concentration of sulphuric acid in exactly the same way and the rates similarly established. When these rates were compared there was no relation whatever between them. Some specimens that had corroded at about the same rates in distilled water corroded quite differently in the acid solution. As this was not the test wanted it was discarded. The acid corrosion test may be of value for certain specific uses, but it certainly can not be generalized from as an accelerated test.

CORROSION OF IRONS, STEEL, AND SOME NON-FERROUS METALS IN DISTILLED AND IN SEA WATERS.

These tests were made in glass jars, some sealed for a year, then left open until water was all evaporated and the pieces were dry, and others were left open and loss from evaporation made up from time to time by addition of water from the same

source. The units are losses in milligrams per square inch of area in times as designated.

Polished boiler-plate steel in distilled water, jars open.

First 30 days,	12.1	First year,	186.0
Fifth 30 days,	17.0	Second year,	158.5

The products of corrosion were left in the jars and losses from evaporation were made up when surface of water was one inch above the upper surface of the steel. This indicates that the presence of the products of corrosion does not accelerate corrosion, but rather that corrosion is retarded thereby.

Steel from the same plate and under the same conditions as the foregoing, with mill scale on top and bottom, the sides and ends being planed:

First 30 days,	12.1	First year,	207.0
Eleventh and twelfth 30 days,	19.9	Second year,	182.4

At the end of the second year the mill scale had almost all been changed to ordinary rust, and the appearances of pieces in this and the preceding tests were very much the same. The rates of loss for the last three periods of the second year were very nearly the same, 11.0 and 12.9, respectively. The pieces were turned over after each weighing, as the bottoms of the pieces next the glass did not lose their polish or mill scale, and the attack was principally on the top, sides and ends. On a polished piece, with the same area always kept on the bottom, most of the polish remains after two years. This may be due to the light being kept away from the bottom of the specimen, or to the fact that the products of corrosion cannot diffuse away and be oxidized, and therefore that the potential of the solution is raised to that of the steel with which it is in contact before sufficient metal has been dissolved away to become noticeable. The latter is the writer's belief.

Polished steel, same conditions as the foregoing, except placed under and in contact with a sheet of copper of similar dimensions:

First 30 days,	17.9	First year,	194.4
Third 30 days,	18.7	Second year,	192.0
Eleventh and twelfth 30 days,	14.7		

The tops in contact with copper and the bottoms in contact with glass have much of the polish remaining, most of the attack having taken place on sides and ends.

Steel with mill scale on tops and bottoms, with sides and ends planed, placed under copper:

First 30 days,	26.0	First year,	218.2
Eleventh and twelfth 30 days,	14.8	Second year,	180.0

Nearly all of the mill scale remained on tops and bottoms.

With the same steel under sheet brass, instead of under copper, very similar results were obtained.

Steel with mill scale under Muntz metal:

First 30 days,	28.4	First year,	283.6
Second 30 days,	29.5	Second year,	236.4
Eleventh and twelfth 30 days,	21.7		

Same as the foregoing except under Tobin Bronze:

First 30 days,	21.5	First year,	238.0
Second 30 days,	26.0	Second year,	222.0
Eleventh and twelfth 30 days,	17.5		

Similar specimens under rolled Monel metal, with rolling finish on tops, bottoms and ends of Monel:

First 30 days,	16.9	First year,	233.4
Eighth and fifth 30 days,	23.8	Second year,	235.2
Eleventh and twelfth 30 days,	19.2		

From an analysis of the above conditions it will be seen that corrosion has decreased with time in nearly every case on steel immersed in its own corrosion products; that the surfaces of steel in contact with some other surface does not corrode in the same way as on surfaces exposed to water alone, even when the metal in contact with it is much lower in potential than itself; that the surfaces exposed to water alone are more severely attacked when the specimen is in contact with a metal of lower potential; and that the attack depends upon the perfection of contact with the metal of lower potential. This

was indicated in the tests with rolled Monel, Muntz metal and Tobin Bronze, which specimens were much thicker and, accordingly, heavier than the copper and brass plates; the pressure was greater and therefore the contact was better.

The following steels and irons in distilled water alone corroded in milligrams per square inch per thirty days' intervals as given. All pieces were planed all over, jars were sealed for first year, then left open until pieces were dry.

	First year.	Subsequently, while evaporating to dryness, jars open.
Class A steel boiler plate,	15.3	21.9
B steel boiler plate,	14.1	22.9
C steel boiler plate,	13.7	21.9
American ingot iron,	13.8	26.4
Cast-iron, Navy specifications,	16.2	27.5
ammonia fittings,	17.6	26.8
machinery castings,	15.9	27.5
grate bars,	16.1	28.4

This is given to show the effect of increased air or oxygen. With the jars sealed there is not enough oxygen passed into the water to oxidize the dissolved products as fast as they can go into solution. The potential of the water is raised by the dissolved and unoxidized particles of iron in it. The difference of potential between the metal and the metal water solution is decreased and the rate of corrosion is decreased. With the jars open, oxygen is admitted, and the rate of corrosion is increased. An increase in the area of the water in contact with air up to a certain limit will increase the rate of corrosion of the metal.

The irons and steels that have shown themselves to be permanently of higher potentials than other irons and steels by the potentiometer measurements, Ferroxyl mount, and by an alkaline concentration test, to be explained later, always have their rates of corrosion increased and that of the metal of lower potential decreased when they are connected by a metallic contact in distilled, brackish, or sea water. This does not mean that the metal of lower potential does not corrode, but

that its rate of corrosion is decreased. Parasitic corrosion is going on over its surface at the same time that it is acting to increase the rate of corrosion on the piece of higher potential.

In no case has a piece of iron or steel, connected by metallic contact to new zinc of the same size, corroded less in a period of thirty days, than a piece of the same steel placed near it in the same water and not connected by metallic contact to the zinc.

When three grades of nearly pure irons, four grades of cast irons and two grades of boiler steel were all connected to one grade of class B steel boiler plate, only two grades of cast irons reduced the rate on the steel below that obtained when pieces of the same steel were connected in pairs.

With the nickel bronzes the rate of corrosion on steel connected by metallic conductors to them, was generally less than that on the same steel connected to the zinc-copper and tin-copper alloys. Also, similarly, the greater the percentage of nickel in the alloy, the less the rate of corrosion on steel similarly connected thereto. Pure nickel is less corrosive to steel than is nearly pure copper under the same circumstances.

Taking the rate of corrosion of boiler-plate steel in sea water when jars were closed as 100, the same rate with jars open was 122.6. The rates in wrought iron and nearly pure iron under similar circumstances were 73.4–107.7 and 87.7–137.5, respectively. With the same three metals placed under and in contact with Manganese Bronze the rates were 112.3–135.9; 111.8–139.0; and 108.2–140.5, respectively. With the same steel under Manganese and Phosphor Bronzes and Composition G, the rates were 112.3–135.9; 109.2–127.7; and 109.2–131.3, respectively. These differences are no greater than will be found between specimens of the same plate immersed alone in distilled water. In analyzing the foregoing it is seen that wrought iron and nearly pure iron are acted upon very differently from steel by a reduction of the oxygen supply, that under the same Manganese Bronze they are corroded to nearly the same amount, and with the one steel similarly placed under different bronzes its rate of corrosion

is very nearly the same. It is well to note here that the two metals must be in metallic contact, otherwise each metal will corrode as if alone in the water, even when they are very close together. The effect one metal will have upon another depends more upon the perfection of the contact than upon any other one thing. Copper in surface contact with steel in distilled water increases the corrosion of steel from 8–12 per cent., while with the two metals screwed onto the ends of an iron wire, the rate is increased from 79–80 per cent.

With non-ferrous metals and cast irons alone in sea water the rates of the losses with jars sealed and open, respectively, in milligrams per square inch, for 30-day periods, were as follows:

	Jars sealed, 12 periods.	Jars open, 3 periods.
Copper,	1.5	3.0
Victor silver,	plus 1.5	minus 3.6
Phosphor bronze,	9.9	15.1
Composition G,	3.8	4.8
Muntz metal,	1.3	4.0
Manganese bronze,	1.7	2.2
Rolled Monel bar,	1.7	1.0
Cast-iron, Naval specifications,	20.9	25.0
ammonia fittings,	18.4	22.7
machinery castings,	21.3	25.9
grate bars,	20.7	22.1

Note the great rate of corrosion on the Phosphor Bronze in comparison with the others. It is almost half as corrodible as cast iron with jars sealed, and more than half with them open.

With boiler plate steel in distilled and in sea waters containing graphite, pieces covered with graphite lost from 3 to 4 times as much per square inch of area exposed as did similar pieces placed in the same water without graphite. The increased rate of corrosion depended to quite a large extent upon the percentage of area covered and also upon the purity of the graphite.

Fig. 1.—Steel Boiler Direct-Tube Type, Copper Tubes. Removed From Captured Spanish Launch in 1899. Safety Valve Set at 200 Pounds.

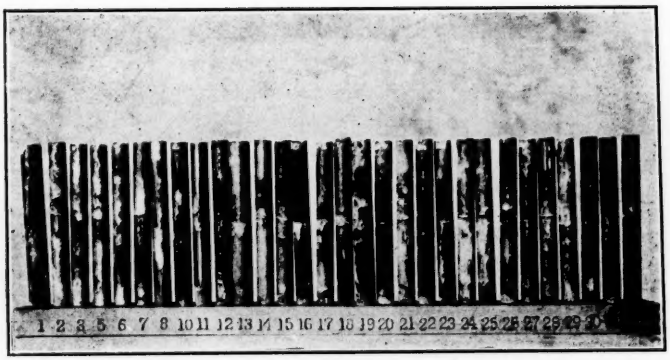


Fig. 4.—CONDENSER TUBES CORRODED SIX MONTHS IN BOX. VULCANIZED RUBBER GASKET AT EACH END TO INSULATE TUBES FROM BOX AND FROM EACH OTHER.

1-3. Cupro nickel tubes.
5-8. Benedict nickel tubes.
10 12. Monel metal tubes.
13-15. Benedict Burnhams tinned tubes.
15-32. Admiralty and Muntz metal tubes, untinned.

Note corroded areas at ends when under rubber.

With pieces of steel immersed in water condensed from exhaust steam from an engine using different kinds of cylinder lubricants (the engine using both saturated and 600 degrees F. superheated steam at 300 pounds gage pressure), there was practically no difference during a twenty months' test in the rate of corrosion as compared with that obtained in steam condensed under similar conditions except free from oil.

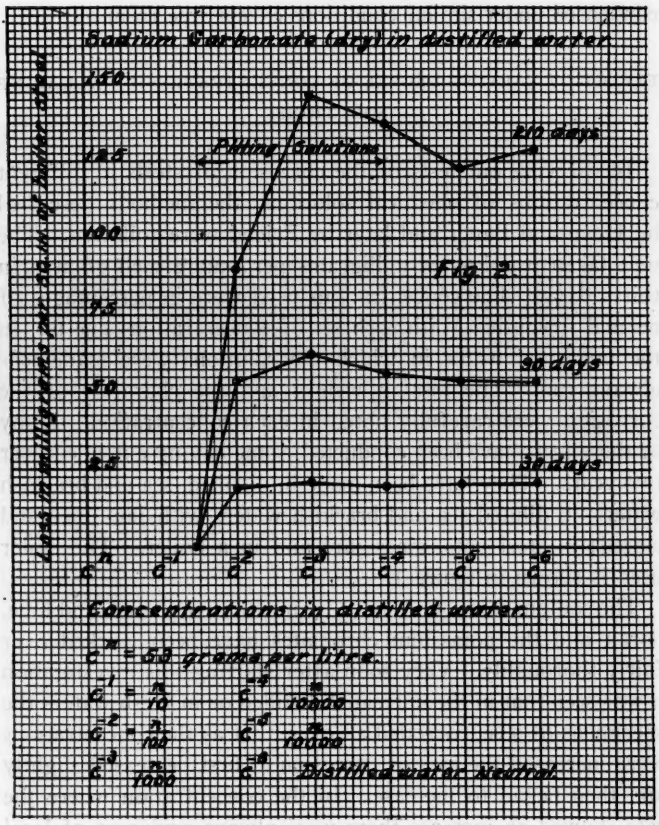
EFFECT OF VARYING DEGREE OF CONCENTRATION OF ALKALINE SOLUTIONS.

In carrying out tests of the corrosion of steels in different concentrations of chemical solutions, the methods described by Heyn and Bauer and later by Friend were followed. The unit taken was loss in milligrams per square inch of area in thirty days. The concentrations used were in percentages of normal strength. With each set of concentrations one test in the untreated water in which the solutions were made was always run. The loss in weight and the character of the corrosion were carefully noted in each concentration. Many of these tests have been conducted, some having extended over two years. They have been made in distilled, sea and brackish waters, in diluted sea and brackish waters with jars open and closed; in steel pots, glass jars and in a boiler. The boiler used is shown in figure 1.

The sodium compounds have been extensively used, and lime, borax and the potassium chromates have been worked over fairly thoroughly. Very soon after starting this investigation it was seen that in the untreated distilled or in fresh waters the specimens of steel corroded all over, that as the concentrations were increased the rate of corrosion at first decreased slightly and then began to increase. In the concentration where the rate showed an increase there were always evidences of local corrosion, or pitting, and as the concentrations increased the rate of loss increased to a maximum, then fell rapidly to zero and remained there, the areas of local corrosion becoming

smaller and more pronounced until they disappeared altogether in the concentration in which the loss was zero. See figure 2.

For the purposes of this article, the concentration whose strength is just below the one in which local corrosion or pitting first appears, will be called the lower-limit concentra-



tion, the one in which the rate of corrosion is a maximum is the critical, and the one just above the one in which the last signs of corrosion appear is the upper-limit one for the metal

tested. The upper-limit concentration in distilled water solutions has been determined for steels with dry sodium carbonate, caustic soda, disodium phosphate, sodium acetate, sodium and potassium chromates and dichromates, sodium silicate, borax and lime. No upper-limit concentration of the chromates, dichromates, and lime in waters containing over 25 grains of chlorine per gallon could be found. The curve for lime is almost flat up to the concentration of saturation, and those for the chromates fall rapidly at about the upper limit in distilled water for the same steel, then rise again with pronounced local corrosion on each slope of the curve.

No upper-limit concentration of sodium in the compounds of the stronger acids could be located, though on some grades of steel the lower-limit and critical concentrations were indicated in the sodium nitrates, chloride and sulphate solutions.

The order of magnitude in which corrosion occurs in distilled-water solutions in the normal concentrations of sodium nitrate, chloride and sulphate is indicated to be in about the order of the strength of the acid radicals.

The upper and lower-limit concentrations in those solutions in which they are found vary with the metals immersed, and with the conditions of the surface of the metal in contact with the solution, both with regard to impurities in it and to the physical treatment it has received.

The rate of loss in the critical concentration is greater than that in the untreated water. No critical concentrations were located in sea water, their locations becoming more apparent as the water became more diluted.

The upper-limit concentrations of the chromates were about the same as those found by Heyn and Bauer and by Cushman, variations being due, of course, to variations in quality and condition of the metals used.

The limit and critical concentrations of the following solutions for one grade of steel have been very carefully determined and are given in percentages of normal strength in distilled water.

Solution.	Upper limit.	Critical.	Lower limit.
Sodium carbonate (calcined),	2.6	0.8	0.16
Caustic soda, ...	2.6	0.8	0.16
Lime, ...	2.6	0.6	0.15
Disodium phosphate, . . .	4.0	1.2	0.16
Sodium chromate, ...	0.26	0.08	0.03
Potassium chromate, . . .	0.26	0.08	0.03
Sodium dichromate, . . .	0.26	0.08	0.03
Potassium dichromate, . . .	0.26	0.08	0.03

Heyn and Bauer did not find critical concentrations in the chromates, dichromates and lime solutions, but they were located here in every case.

In further work along these lines with open jars it was seen that the concentration of lime solutions weakened very rapidly while those in caustic soda, sodium carbonate, disodium phosphate and the chromates maintained or increased their concentration as the water was evaporated away. Solutions of lime water exposed to the air may be non-corrosive to steel one day and corrosive the next, due to the absorption of carbonic acid from the air, while if the upper-limit concentration of sodium carbonate is made, steel thrown in it will not corrode until the concentration is brought below the upper limit by some external means.

EFFECT OF TEMPERATURE ON CONCENTRATION LIMITS.

No concentration that has been found to be noncorrosive to iron or steel at room temperatures has ever been found corrosive at temperatures up to 420 degrees F., that corresponding to a steam pressure of 300 pounds gage. It has been found that concentrations just below the upper limit, and therefore corrosive, were noncorrosive to the same steel at higher temperatures. It must be that with rise in temperature the potential of the solution rises faster than that of the metal. By a long series of experiments it was found that steel connected to copper by a good metallic conductor does not corrode in an upper-limit concentration of sodium carbonate, lime, caustic

soda, or disodium phosphate. At the first signs of corrosion as shown by reddish brown or black discoloration, the concentration was determined by titrating with acid and was always found to be below the upper limit for that for steel or iron. In similar concentrations *air, oxygen, carbonic acid gas, graphite, zinc oxide, mill scale, and other supposed exciters of corrosion have no effect upon steel or iron until the concentration is reduced below the upper limit for that steel.* Steel was suspended in a glass basin in, and above, the limit concentration of sodium carbonate, and pure oxygen was blown under it in such a way that sixty bubbles a minute impinged on the steel and passed up along its sides; this continued for eight days, and no signs of corrosion were evident and there were no losses of weight. This was continued fifteen days longer, using air instead of oxygen, with the same result.

Zinc, being of a higher potential than iron, will corrode in an upper-limit concentration for iron or steel.

It was found that the *upper-limit concentrations for all irons and steels tested was about 2.6 per cent. normal alkaline strength of sodium-carbonate and of caustic-soda solutions, in distilled and in sea water.* The highest concentration of any of these chemicals in which corrosion has been found is 2.5 per cent. of normal.

It must be understood here that these concentrations are made by adding the chemical and titrating the solution with acid, and not by adding chemical by weight alone. It takes much more soda to bring the concentration of one gallon of sea water up to 2.6 per cent. normal alkaline strength than it does for the same amount of distilled water.

THE CAUSE OF THE CORROSION OF THE "NEW JERSEY'S" BOILERS.

While carrying out these tests it occurred to the writer to calculate the approximate concentration of lime that had been kept in the *New Jersey's* boilers. This was found to have been at about the critical concentration for boiler steel, so that in trying to stop corrosion by the use of lime, zincs and other

methods, local corrosion or pitting had been induced. Had twice as much lime been used, no local corrosion would have occurred even with the composition liners and air. To prove this, the small steel boiler, with copper tubes, shown in figure 1, was run for 15 days at a saturation of 2,000 grains of chlorine per gallon, keeping the water at a concentration of 3 per cent. normal with soda. Steam pressure carried was 180 pounds gage. At the end of that time five test pieces placed as follows showed no signs of corrosion and no loss of weight:

One at bottom of shell at smokepipe end,
One at bottom of shell at furnace end,
One on top of crown sheet,
One laid across copper tubes at middle of their height, and,
One laid across copper tubes near the water level.

The boiler was fed with brackish water from the Severn River, and was given a blow down when the chlorine content was found above 2,200.

It thus was demonstrated why the boilers of the *Oregon* showed no signs of corrosion in three years, and why those of the *New Jersey* were considerably corroded in the same time. In the one case corrosion had been prevented by keeping the water strongly alkaline, in the other it had been materially aided by an insufficient degree of alkalinity.

In calculating the percentage of soda concentration kept in the boilers of the *New Jersey* it was seen that it was slightly lower than the critical, but still a dangerous one to use. The reason why there was no corrosion before the dynamos of the *New Jersey* were raised was that there was always a film of oil between the metal and the water, so no bad effects from corrosion could occur. Oil on the surface of water passes oxygen to it and thereby aids the corrosion by removing the dissolved particles of iron from solution, but oil on the surface of the metal prevents water from getting to it and thereby prevents corrosion so long as the skin of oil is intact. As soon as the access of oil to the boilers was prevented, the tubes

began to pit and corrosion began. It was only partly due to any particles of composition, partly to the graphite from the gaskets, but principally to the water conditions and to zinc scale. *Had the water been kept properly alkaline the other three conditions would have had no effect.*

The writer has no hesitancy in saying that *any boiler using any water can be kept from corroding for any length of time,* if *treated with soda, and if its concentration is maintained at or above 3 per cent. normal alkaline strength. If the water is not to be kept sufficiently alkaline, it had better be kept neutral.*

INDICATORS.

The subject of indicators is a complex one, and the determination of neutrality of water is difficult. Litmus does not indicate neutrality correctly unless the water is boiled, phenolphthalein does not indicate the neutral point when there is ammonia in the water or when it contains carbonates, and practically none of the indicators shows fatty and the weaker organic acids. Boiler water may be above the upper-limit concentration of alkalinity due to bicarbonate of soda and not be indicated that it is alkaline by phenolphthalein. The only way in which boiler water can be kept at the proper concentration of alkalinity is by drawing water from the boiler and titrating it with a strong acid, understanding the action of the indicator used, or by maintaining a representative sample of the boiler water in a glass jar in which is suspended a clean piece of boiler tube. Soda is added to the boiler water whenever corrosion is seen to be taking place in the sample jar within from 8 to 12 hours. There can be no thumb-rule methods of pounds of soda per I.H.P., per pound of coal burned, or per gallon of feed water. The water in the boiler must be kept at, or above, a certain concentration of alkalinity, and that can only be determined from the water that is in the boiler. Each boiler must be considered as a separate unit.

After a boiler has been pumped up to steaming level with water of proper alkaline strength, and steam is raised, the concentration may be reduced as follows:

(a) Some soda will be taken over with the steam if the boiler primes.

(b) Soda will be lost in any water that may leak from the boiler or be blown out of it by the surface and bottom blows.

(c) Soda will be lost by combining with some of the impurities that come in with the feed water.

If the boiler is perfectly tight, the blows are not used and only pure feed water is fed to it and the water level is kept constant, the concentration of soda at the end of any run should be the same as that at the beginning if the boiler is delivering dry steam.

BOILER COMPOUNDS.

Sodium carbonate, caustic soda, and lime, added to boiler water, tend to make it foam or prime, and if the water is not pure they form precipitates with some of the impurities which are thrown down on the heating surfaces and form hard scale. There are other chemicals that can be added to the water, such as sodium phosphate, that reduce this tendency to priming and still others like tannic acid, deatrine and others that hold the precipitates in suspension and prevent their forming scale. They form a sludge of the scale which can be blown out through the bottom blows.

With pure feed water, sodium carbonate and disodium phosphate, properly proportioned, will stop corrosion and priming if enough of the mixture is used. If impure water is used, then sodium carbonate, disodium phosphate and cutch (containing tannic acid) will, when used in the right amounts, stop corrosion, prevent priming, and also prevent scale from forming unless the saturation of sludge gets too high. Such a mixture is the Navy Standard Boiler Compound, and if the water in the boiler is, by use of this compound, always kept at a concentration of or above 3 per cent. normal alkaline strength, no corrosion will take place, no scale will form and the water will be no more likely to prime than it would be if it were untreated.

This is a powdered compound, composed of calcined sodium

carbonate, trisodium phosphate, dextrine or starch, and a tannin compound, as mangrove bark, cutch, or catechu. These materials are intimately united by thorough digestion, dried, finely powdered, well mixed, and readily soluble in water. The compound must show on analysis at least 64 per cent. of anhydrous sodium carbonate (Na_2CO_3), 15 per cent. of trisodium phosphate ($Na_3PO_4.12H_2O$), 1 per cent. of dextrine or starch, and 10 per cent. of tannin compound containing at least 35 per cent. of tannic acid, the remainder to consist of water and only such impurities as are common to the ingredients.

To determine the sodium carbonate, incinerate 1 gram, dissolve in water, and wash into a flask. Add an excess of standard acid, boil, and titrate back with standard alkali, using phenolphthalein as indicator. Calculate to Na_2CO_3, which result is the actual sodium carbonate plus the sodium carbonate equivalent of the alkalinity of the trisodium phosphate. Therefore deduct from this figure $^{14}/_{100}$ of the percentage of fully hydrated trisodium phosphate found present, and the result is the percentage of sodium carbonate.

The writer has been testing boiler compounds for over two years, has found some good, when used properly, and some bad under any circumstances. They are all bad when used improperly.

ACCELERATED CORROSION TESTS.

If the upper and lower-limit concentrations in sodium-carbonate solution of several grades of irons and steels are determined carefully, it can readily be determined which sample will corrode at the greater rate under given conditions. The one which has the greater range of concentration between its upper and lower limits will corrode the faster. The one having the highest upper-limit concentration will have its own rate increased and that of any other one decreased when connected by good metal contact with it. It is believed that a reliable accelerated corrosion test can be made, using different concentrations of soda or of any other metal that is higher in the electromotive series than iron and which will remain in solu-

tion in water. The writer has used this test for some time and has come nearer to getting concordant results with it than with any other accelerated test of which he has any knowledge. The reason that absolutely concordant results have not always been obtained is believed to be that any corrosion test deals only with the metal in the exposed surface. When that surface is removed, by solution, the new surface exposed may have different characteristics from that just removed. By the test under discussion the writer has found that some boiler plates are of higher potential at the center of the thickness than at the rolled surfaces, probably due to effects of work in rolling or to stresses from cooling.

AN ELECTRICAL PROCESS FOR PREVENTING CORROSION.

Experiments with the Cumberland electrical process of preventing corrosion in boilers, condensers and piping have shown it to be practicable and effective when carefully attended. The diagrammatic arrangement of this installation is shown in figure 3. There is required a current from a source independent of the main electric plant, and of proper E.M.F., kept on at all times. Electrodes are suspended in the water in the boiler and insulated from the metal of the boiler. These electrodes are the positive pole of a battery, of which the boiler material is the negative pole. The potential of the boiler water is raised by the particles of the dissolved anode to a point when it is higher than that of the metal of the boiler. This process has been tried by the writer in the boiler shown in figure 1 when using distilled, well, Severn River, and distilled water made 1 per cent. normal acid strength, without signs of corrosion, either on the boiler metals or on test specimens placed in it.

This process requires constant care. It must be so run that the electric potential of the water in the boiler is kept at a higher point than that of the materials it is designed to protect over every point in their exposed surfaces.

Particular care must be taken to prevent a reversal of current, which, of course, will induce rapid corrosion.

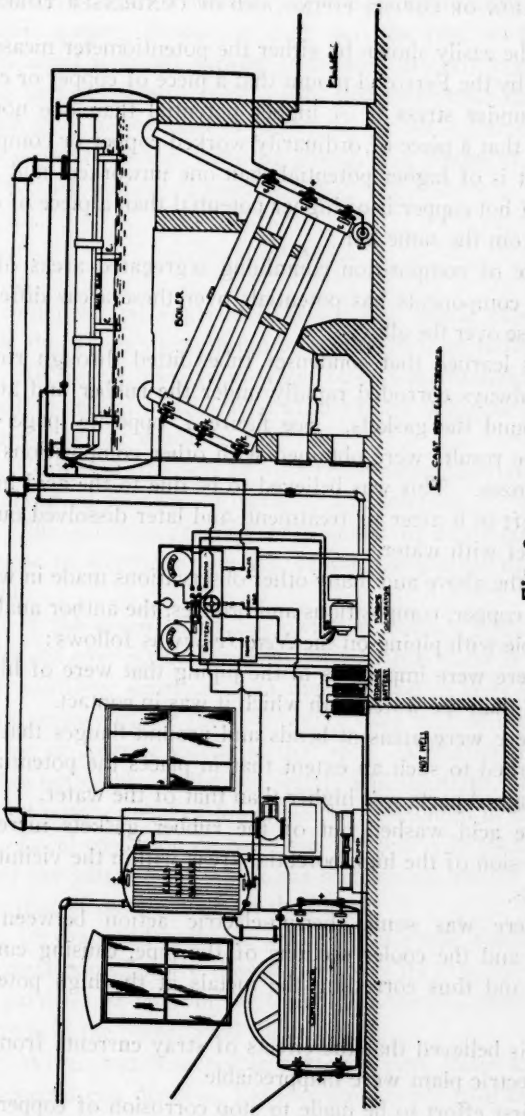


Fig. 3.

ELECTRICAL PROCESS FOR PREVENTING CORROSION IN BOILERS, CONDENSERS AND PIPING.

CORROSION OF COPPER PIPING AND OF CONDENSER TUBES.

It can be easily shown by either the potentiometer measurements or by the Ferroxyl mount that a piece of copper or composition under stress is of higher potential than one not so stressed; that a piece of ordinarily worked copper or composition sheet is of higher potential than one unworked, and that a piece of hot copper is of higher potential than a piece of cold copper from the same bar.

A plate of composition containing segregated areas of its different components has potentials over these areas different from those over the alloy.

It was learned that condenser tubes fitted through rubber gaskets always corroded rapidly under the rubber and at the ends around the gaskets. See figure 4, opposite page 859. The same results were obtained with other compositions and with bronzes. This was believed to be due to the acid in the rubber left in it after its treatment, and later dissolved out by its contact with water.

From the above and many other observations made in working with copper, compositions and bronzes, the author analyzes his trouble with piping on the *New Jersey* as follows:

1. There were impurities in the piping that were of higher potential than the water with which it was in contact.

2. There were areas at bends and around flanges that had been worked to such an extent that in places the potential of these strained parts was higher than that of the water.

3. The acid washed out of the rubber gaskets increased the corrosion of the high-potential areas within the vicinity of the joints.

4. There was some thermo-electric action between the warmer and the cooler sections of the pipe, causing current to flow and thus corroding the metals at the high potential points.

5. It is believed that the effects of stray currents from the ship's electric plant were inappreciable.

The first effort to be made to stop corrosion of copper and composition piping should be in the direction of getting metal with the same chemical structure throughout. All parts should

be annealed to give a uniform physical structure after working, metallic gaskets should be used, and the pipe so built that when fitted it is not sprung or twisted in such a way as to bring unequal strains in it. Any thermo-electric effects will either have to be taken or cared for by use of zincs kept in perfect metal contact with the piping, and placed at least one zinc to each section of pipe.

SUMMARY.

From practical experience, study, and much experimental work, the writer's conception of corrosion is as follows:

1. All metals dissolve in water or in water solutions if the electric potential of the metal at any point is higher than that of the water or solution.

2. Corrosion products or rust result from the oxidation of the dissolved particles of metal held in solution.

3. The solution pressure of a metal in any liquid is the increase in electrical potential of the metal over that of the liquid.

4. If the potential of the metal in contact with water is higher all over its surface than that of the water it will corrode evenly or nearly so. If the potential of only one point in the surface of the metal is higher than that of the water it will pit or corrode locally at that spot. Metals do not pit or groove noticeably in distilled water, acid solutions, or when in contact with a metal of lower electrical potential in water.

5. Local corrosion on bare iron or steel only takes place in water the potential of which has been raised until it is higher than that of certain areas of the metal.

6. Oxygen and carbonic acid do not cause corrosion. If the metal can dissolve they do increase its rate by removing the particles of it from the liquid and thereby preventing an increase in the potential of the liquid.

7. When iron is placed in perfectly pure water into which no oxygen can dissolve, the iron first tends to dissolve, due to its higher potential. Each particle of iron dissolved raises

the potential of the water and exposes a new particle of iron. Eventually the potential of the water will be equal to that of the metal at every point, dissolving will stop and the solution will be saturated. Now if oxygen is present in the water, as the iron is dissolved it will be oxidized, the potential of the liquid will be raised only to the point where there is equilibrium between the rate of solution of the iron and that of the oxygen, and corrosion of the iron will continue.

8. Metals corrode in sea water at a greater rate than in distilled water due to the higher conductivity of the former. The metal particles get away from the metal and in contact with oxygen or CO_2 more freely and the potential of the water is kept from rising. The natural potential of sea water is not raised to any appreciable extent by the metallic salts it contains, because the high-potential metals of those salts are mostly combined with strong acid radicals such as chlorides and sulphates.

9. Hydrogen and all elements lower in the electromotive series than hydrogen decrease the potential of water in which they are in solution. All elements electropositive to hydrogen increase the potential of the water when they are in solution.

10. The alkalies are the highest in the electromotive series, and are therefore the most soluble in water. They raise the potential of water in which they are in solution.

11. The sodium salts of weak acids increase the potential of water at a greater rate than do the same salts of strong acids.

12. The potential of a solution containing a metal higher in the electromotive series than another metal immersed in it will increase faster with a rise in temperature than that of the immersed metal.

13. When boiler water is properly treated, air, copper, graphite, zinc oxide and other electronegative metals in contact with iron or steel have no effect whatsoever in starting and maintaining corrosion, but they do have a very serious effect if the water is not properly treated, and their effect increases with the perfection of their metallic contact with the metal of the boiler.

THE NAVY STANDARD BOILER WATER-TESTING OUTFIT AND INSTRUCTIONS FOR USE.

DESCRIPTION OF OUTFIT.

1.* Burette in place ready for use for determination of alkalinity. Graduated to $\frac{1}{10}$ c.c., at 15 degrees C., 50 c.c. capacity.

2. White porcelain dish of 200 c.c. capacity.

3. Bottle of one-tenth normal sulphuric acid. One-tenth normal H_2SO_4 contains 4.9 grams of pure acid in a liter of solution in distilled water. (See par. 9, under "General Instructions.")

4. Measuring cylinder graduated to 1 c.c., at 15 degrees C., 100 c.c. capacity.

5. Burette, similar to 1, for determination of chlorine content.

6. Spare burette, similar to 1 and 5.

7. Glass rod for stirring samples in 2.

8. Pipette of 1 c.c. capacity for dropping indicator, etc., in sample and for general measuring adjustments.

9. Bottle of standard neutrality indicator, fitted with dropper through stopper, 100 c.c. capacity. The indicator is made by dissolving 1 gram of methyl orange powder and 5 grams of phenolphthalein powder in 500 c.c. of alcohol and adding enough distilled water to make a liter of solution.

NOTE.—This indicator, if added to an acid solution, will turn it pink, due to the methyl orange in the indicator. If added to an alkaline solution it will turn to a deep red, due to the phenolphthalein in the indicator. If the solution is exactly neutral the color will be a faint yellow, due to the methyl orange.

10. Bottle of normal solution of caustic soda 500 c.c.

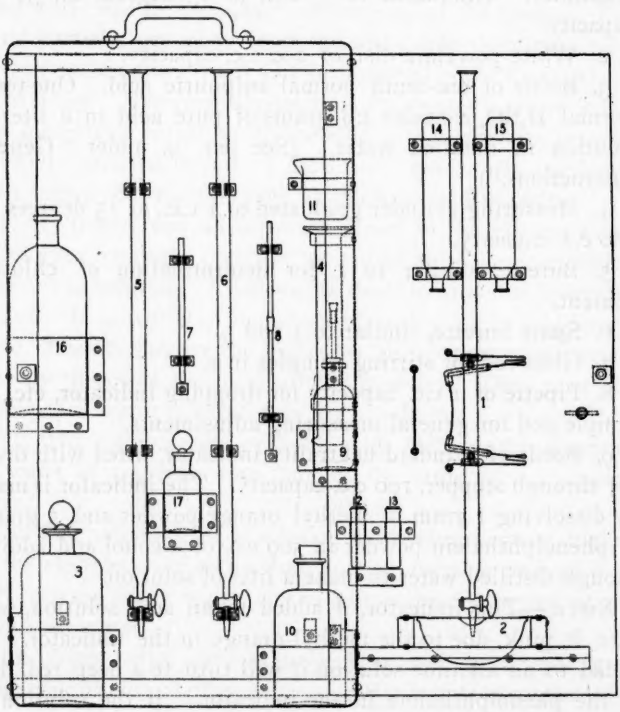
* Numbers refer to accompanying illustration.

capacity. The solution is made by dissolving 40 grams of pure caustic soda (NaOH) in 1 liter of solution in distilled water.

11. Glass beaker 150 c.c. capacity for pouring acid, alkali, or nitrate of silver solutions into burette.

12. Bottle of red litmus paper strips for general use in testing strong alkalies. Strong alkalies turn the red paper blue.

13. Bottle of blue litmus paper strips for general use in testing strong acids. Strong acids turn blue paper red.



14 and 15. Graduated test tubes for measuring large amounts of chlorine in water and for diluting water for chlorine measurements. The graduations are in 5 c.c. increments, but are marked 0, 50, 100, etc., as in the usual B. & W. measuring bottle.

16. Bottle of nitrate of silver solution, 1 liter capacity. The solution is made by dissolving 4.101 grams of chemically pure nitrate of silver ($AgNO_3$) crystals in a liter of solution in distilled water.

17. Bottle of potassium chromate indicator, 100 c.c. capacity. The solution is made by dissolving 1 gram of potassium chromate (K_2CrO_4) in 100 c.c. of solution in distilled water.

METHOD OF USING.

1. *Alkalinity Test.*—Draw a sample of water from the boiler into a glass or porcelain receptacle *which has just previously been washed out with water from the same boiler.*

2. Fill the burette 1 with acid from bottle 3, using beaker 11. Open pet cock at bottom of burette and draw a few drops of acid through it into 11. Repeat this, if necessary, until all air bubbles have been expelled from lower end of burette and it is filled to tip with acid when cock is closed.

3. Measure exactly 50 c.c. of the sample of boiler water into cylinder 4, and pour it into dish 2, which has just previously been washed out with other water from the same sample, or with distilled water, and wiped dry.

4. Drop 4 drops of neutrality indicator from bottle 9 into sample in dish 2. The sample, if alkaline, will now turn a deep red color, due to the reaction between the phenolphthalein in the indicator and the carbonate or caustic alkali in the water.

NOTE.—Should the sample be alkaline, due to the presence of a bicarbonate, the reaction will not take place. This, however, is a condition not likely to exist when Navy standard boiler compound or plain carbonate of soda or a combination of the two is used in the boilers.

5. Read the graduation at the top of the acid in the burette; then from pet cock at the bottom drop acid into the sample in dish 2, stirring continuously with the glass rod 7 until sample turns clear, then yellow, then a very faint pink. Close the pet cock and read the graduation on the burette at

the top of the acid. The difference between the two readings indicates the number of c.c. of acid required to neutralize the alkali in the sample. The percentage of normal alkaline strength, X, of a solution may be expressed thus:

$$X = \frac{A}{S} \times P,$$

where A equals number of c.c. of acid required exactly to neutralize the alkali in the sample; S equals the number of c.c. of the original sample, and P equals the percentage of normal strength of the acid used.

For example, if in the above test it required exactly 4 c.c. of the one-tenth normal acid solution from the burette to neutralize the alkali contained in the 50 c.c. of the sample in dish 2, the percentage of the normal alkaline strength of the sample is

$$\frac{4}{50} \times \frac{1}{10} = \frac{4}{500} \text{ or } .008;$$

in other words, the sample is eight-tenths of 1 per cent. of normal alkaline strength.

To be safely noncorrosive the boiler water must be at least 3 per cent. of normal alkaline strength. Hence, if less than 15 c.c. of tenth normal acid is required to neutralize the 50 c.c. sample of boiler water, the latter is not safely noncorrosive, and additional compound is required in the boiler.

NOTE.—These instructions require acid to be added until the sample turns a faint pink. This indicates that the alkali has been a little more than neutralized and the sample has become slightly acid, and consequently the sample is not as strongly alkaline as the formula would indicate. The error, however, will be negligible for the purpose of testing boiler water if the test is carefully made and the pet cock closed at the instant that the faintest pink color is attained in the well-stirred sample. For this reason the acid should be added drop by drop and the test conducted in a good light.

6. *Chlorine Test.*—The sample in dish 2 is now slightly

acid. Before testing for chlorine the sample must be made neutral or slightly alkaline.

Using pipette 8, drop caustic-soda solution from bottle 10 into the sample until it becomes slightly yellow, indicating that it is neutral; or red, indicating that it is alkaline.

7. Replace burette 1 with burette 5, and fill 5 with silver-nitrate solution from bottle 16, taking precautions as before to see that it is filled to the tip. Using pipette 8, drop 4 drops of chromate indicator from bottle 17 into sample in dish 2. Read the graduation at the top of the nitrate in the burette, then from pet cock drop nitrate into the sample in dish 2, stirring continuously with the glass rod 7 until the sample turns a reddish-yellow color throughout. Close the pet cock and read the graduation on the burette at the top of the nitrate. The difference between the two readings indicates the number of c.c. of the nitrate required to precipitate all of the chlorine in the sample.

When any volume of solution containing 4.101 grams of silver nitrate per liter is just sufficient to precipitate all the chlorine in an equal volume of a sample, the sample contains 50 grains of chlorine to the gallon of 231 cubic inches, Hence the chlorine content of a sample may be expressed thus:

$$X = \frac{N}{S} \times 50,$$

where X equals the grains of chlorine per gallon of sample, S equals c.c. of original sample tested, and N equals c.c. of nitrate of silver solution of 4.101 grains per liter required to just precipitate all the chlorine in the sample. For example, if in the above test it required 3 c.c. of the nitrate solution from the burette to turn the well-stirred sample of 50 c.c. a reddish-yellow the chlorine content is $\frac{3}{50} \times 50$, or 3 grains of chlorine per gallon of water.

The stopcock should be closed as soon as the change in color from yellow to reddish-yellow in the well-stirred sample occurs. If the nitrate is added until the sample is a deep red, erroneous results will be derived. For this reason the nitrate

should be added drop by drop after the first sign of reaction occurs, and the test should be made in a good light.

NOTE.—The sample in dish 2 at the beginning of the chlorine test consisted of more than 50 c.c. of total contents, being made up of the original 50 c.c. of the boiler water, and the amounts of indicators, acid and alkali added subsequently. However, the reaction for determination of chlorine depends upon the precipitation of all chlorine contained in dish 2, and since none has been either added or subtracted by the addition of the other reagents, the amount of nitrate of silver required to precipitate all the chlorine in the augmented sample is a direct measure of the chlorine content in the original sample of 50 c.c., expressed in terms of grains per gallon.

8. The test described in paragraph 7 will determine the chlorine content within a fraction of a grain, and should be employed when testing water from condensers, distillers and feed tanks, and generally any water known or supposed to contain less than 50 grains of chlorine to the gallon. For the rough determination of high chlorine content the graduated test tubes 14 and 15 may be used as follows:

Pour a small sample into cylinder 4, and by adding one or two drops of the neutrality indicator from bottle 9, determine that it is neutral or alkaline, as indicated by a yellow or red color, respectively. If it is acid, as indicated by a pink color, it must be neutralized by adding caustic soda solution from bottle 10, drop by drop, until the pink color is replaced by yellow or red. Decant into 14 (or 15) until the top of the sample is level with the graduation marked "O." The tube now contains 5 c.c. of the sample. Add one drop of the chromate indicator from bottle 17; slowly add silver nitrate solution from bottle 16; keep shaking the bottle. On nearing the full amount of nitrate solution required the sample will become reddish for an instant, but will turn back to yellow when shaken. Add nitrate solution drop by drop, and as soon as the sample shows a reddish-yellow and remains that color when shaken stop adding nitrate. The reading of

the graduated bottle at the top of the sample will show the grains of chlorine per gallon. This is because the graduations, beginning from the bottom of the bottle, are in increments of 5 c.c., and, as stated in paragraph 7, a given volume of a nitrate of silver solution containing 4.101 grams to the liter will just precipitate all of the chlorine in an equal volume of water containing chlorine in the proportion of 50 grains to the gallon. It follows that double the amount of nitrate solution will be required to cause the same reaction in water containing chlorine in the proportion of 100 grains to the gallon, etc.

Should the sample be supposed or known to be strongly alkaline it must, after the standard neutrality indicator has been introduced, and before the sample is decanted into tube 14, be treated with acid from bottle 3 until it turns faintly pink. Then add one drop of caustic solution from bottle 10, when the pink color will be replaced by either yellow or red. Now proceed as before.

NOTE (*a*).—The fact that one or more reagents must be added to the sample before the silver nitrate solution is added, and the proportional volumes thereby disturbed, vitiates somewhat the accuracy of the results of this method of test. But as the method is necessarily a rough one at best, the error introduced by adding the reagents will be negligible unless a relatively large quantity of acid or alkali has been required to make the sample nearly neutral.

NOTE (*b*).—Using this method, chlorine up to about 900 grains per gallon may be measured. If the sample contains more than this proceed as follows: Take 10 c.c. of the sample and dilute with 90 c.c. of distilled water. Take 5 c.c. of the diluted sample and proceed as before. Each 5 c.c. of silver nitrate added now indicates 500 grains of chlorine to the gallon. Other proportions may be used in a similar manner.

GENERAL INSTRUCTIONS.

9. The accuracy of the results of the alkalinity test depends directly upon the accuracy with which the acid solution in bottle 3 is compounded. A bottle of concentrated sulphuric

acid should be carried in stock, and the solution of $\frac{1}{10}$ normal strength prepared as required in the following manner:

Take about 950 c.c. of distilled water and to it add slowly about 4.5 c.c. of the concentrated acid. (The acid must be added to the water, for if water is added to the acid so much heat may be generated that the mixture will splutter and be thrown out of the receptacle.) To the solution now add sufficient distilled water to make 1,000 c.c. of solution.

Determine the exact acid strength of the solution as follows:

Measure 50 c.c. of the solution as a sample into dish 2. Fill burette 6 down to tip with normal caustic soda solution from bottle 3. Drop 4 drops of neutrality indicator from bottle 9 into the sample, which will turn pink. Read height of caustic soda solution in burette; run caustic soda solution into the sample until the pink color is replaced by a faint yellow, denoting that the acid in the sample has been exactly neutralized. Read the height of the solution in the burette and determine how many c.c. of the caustic soda solution have been used in neutralizing the acid in the sample. Remembering that the caustic solution is of normal strength, and that the acid sample contains 50 c.c., it is evident that if exactly 5 c.c. of the alkaline solution just neutralized the acid, the acid strength is $\frac{5}{50}$ or $\frac{1}{10}$ normal. If it required but 4.5 c.c. of the caustic solution, then the acid strength is $\frac{4.5}{50}$, or 90 per cent. of $\frac{1}{10}$ normal, or $\frac{9}{100}$ normal.

Suppose, in testing a sample of boiler water for alkalinity, it was found that 15 c.c. of the acid solution of the latter strength was required to turn the 50 c.c sample pink. Then employing the formula in paragraph 5,

$$X = \frac{A}{S} \times P = \frac{15}{50} \times \frac{9}{100} = .027,$$ or 2.7 per cent. of normal alkaline strength—not a safely noncorrosive condition.

10. It is necessary after filling a burette with any liquid from the beaker 11 to hold the beaker under the burette, open the stopcock, and let a few drops of the liquid run into the beaker.

This will expel the air; and the tip of the burette below the cock will be found to be filled with liquid when the cock is closed. If the first reading is taken before the tip has been filled with liquid the difference between first and second readings will be in error by the amount of liquid required to fill the tip below the cock, plus the volume of any air bubbles which may be held in suspension above it.

11. It is important that before collecting samples of water from boilers, tanks, distillers, etc., the receptacle in which the sample is collected be well rinsed out with distilled water or water from the same boiler, tank, or distiller which is to be sampled. Otherwise precipitates from a previous sample may remain in the receptacle and be redissolved, and the test will give erroneous results. Similarly, the dish 2 should be thoroughly washed and dried after each test to remove traces of previous sample and of the reagents used in testing it.

12. It is recommended that experiments be made in clear daylight and the colors due to the reaction of the standard neutrality indicator in acid, neutral and alkaline solutions, respectively, be carefully noted, and that these colors be reproduced in water colors on a strip of paper to be posted on the door of the case. This will be found useful as a reference when tests must be made in dim light or artificial light, and will also be found useful as a guide when tests are being made by a novice.

13. The litmus papers are furnished for rough qualitative alkaline or acid determinations. They should be used with caution. When used with water or any other liquid having an affinity for CO_2, the liquid should be boiling, since the presence of CO_2 will cause the color reaction to lag and results may be very misleading.

TEST OF A BLAKE VERTICAL SIMPLEX BOILER-FEED PUMP AT THE NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MARYLAND.

The object of the test was to determine the overall mechanical efficiency, water rate in pounds of dry steam per I.H.P. per hour, slip and duty at different piston speeds, discharge pressures and temperatures of feed water.

DESCRIPTION OF THE PUMP.

The pump is of the vertical simplex, inside-packed type, and is of the following dimensions:

Number of cylinders	1
Diameter of steam cylinder, inches	10
piston rod, inches	2
Area of top end of steam piston, square inches	78.54
bottom end of steam piston, square inches	75.40
Nominal stroke, inches	12
Diameter of water plunger, inches	7
rod, inches	2
Area of top end of water plunger, square inches	35.34
bottom end of water plunger, square inches	38.48
Nominal stroke, inches	12
Water plunger packed with five rings of $\frac{3}{8}$-inch by $\frac{3}{8}$-inch "Aqua" hydraulic packing.	
Number of suction valves	8
Area of suction valves (clear), each, square inches	2.52
Maximum lift, inch	$\frac{13}{32}$
Type	Solid composition.
Number of discharge valves	8
Area of discharge valves (clear), each, square inches	2.52
Maximum lift, inch	$\frac{13}{32}$
Type	Solid composition.
Area of steam pipe, square inches	1.496
exhaust pipe, square inches	2.036
suction pipe, square inches	9.887
discharge pipe, square inches	7.393

The main valve is of the floating type, operated by a pilot valve which controls the motion of the piston carrying the main valve. The pilot valve is actuated by an oscillating rod taking its motion through a link from the crosshead. The pivot of the oscillating rod is formed on a cast-iron bracket bolted to the bottom cylinder head. Two tappets with tapped bushings held fast to the pilot valve rod by set screws shift the valve at the end of the stroke. These tappets allow of a wide range in length of stroke, and possess the further advantage of permitting adjustment while the pump is in operation. Although the valve mechanism consists of a great number of moving parts, the design is such that there is scant liability of their getting out of order.

METHOD OF TEST.

Thirty-two tests were run in all, sixteen with cold Severn River water and sixteen with water at about 150 degrees F. These main groups were subdivided into four groups of four tests each, at piston speeds of 25, 50, 75 and 100 feet per minute and at discharge pressures of 200, 250, 300 and 325 pounds per square inch gage.

The duration of the tests was determined by the rate of discharge. At piston speeds of 25, 50 and 75 feet per minute the hourly capacity was less than the 13,000-gallon tank from which the pump took its suction, so the length of test was fixed as 60 minutes and observations were taken every 15 minutes. At a piston speed of 100 feet per minute the tank was emptied in less than one hour, so the length of test was fixed at 40 minutes and observations were taken every 10 minutes. The water for hot-water tests 17 to 32 inclusive, was heated by admitting steam to the tank from which the pump took its suction.

The exhaust from the pump was piped to a condenser, the steam condensed and weighed in tanks on platform scales, which were calibrated and found correct. The condenser was tested for leaks at different times during the test and found tight.

A valve was located on the exhaust line between the pump and the condenser, by means of which a back pressure of about 5 pounds gage could be maintained on the pump.

The quality of the steam to the pump was computed from observations made with a Carpenter's throttling calorimeter fitted between the throttle valve and the pump.

The I.H.P. was determined by five sets of indicator cards taken at regular intervals during the tests from both the top and bottom ends of the cylinder.

The total travel of the piston was taken by a recording-stroke meter attached to the pump rod, and the number of double strokes by a continuous rachet counter attached to the pump rod. Great care was taken to keep the rate of piston travel uniform throughout the test.

The pump suction was an 8-inch pipe up to within four feet of the pump, where it was reduced to the size of the pump suction. The suction was taken from a 13,000-gallon tank, the bottom of which was about three feet above the suction valves of the pump, thus putting a pressure on the suction side.

The water pumped in all the cold-water tests was measured by the drop in level during the test of the water in the supply tank. This tank had previously been calibrated. The water pumped in hot-water tests at 25 and 50-foot piston speeds (tests 17 to 24 inclusive) was determined by flow through a 3-inch by 1¼-inch Venturi meter, and at 75 and 100-foot piston speeds (tests 25 to 32 inclusive) by flow through a 6-inch by 2½-inch Venturi meter. Both meter tubes were calibrated with hot water at the various rates of flow at which they were used and observations corrected accordingly.

The suction pressure was measured by a mercury gage placed in the suction pipe at the entrance to the pump, while the discharge pressure was measured by a gage, reading in pounds, attached to the discharge chamber of the pump. A 6-inch air chamber, eight feet long, equalized pressure on the suction side and an 8-inch chamber eighteen feet long was attached on the discharge side. To insure the full benefit of the chamber on the pressure side it was charged with compressed air.

Test Number.	Date.March-April.1912.	Duration of Test.Minutes	Barometer,corrected. Inches of mercury.	DATA FROM STEAM END. Steam pressure at boiler. Pounds. Gage.	Steam pressure in steam chest, Pounds. Gage.	Steam pressure in steam chest.Pounds. Absolute.	Back pressure.Pounds. Absolute.	M.E.P. of steam cylinder. Top end.	M.E.P. of steam cylinder. Bottom end.	Piston speed, Feet per minute.	I.H.P.developed by steam cylinder.	Steam consumed per hour. in pounds.	Quality of steam, per cent.	Dry steam consumed per hour in pounds.	Dry steam consumed per minute in pounds.	Total B.T.U.supplied pump per hour.	Total B.T.U.rejected by pump and used in heating feed water, per hour.	Number of B.T.U.consum-
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	N
1	18	60	30.11	319.4	108.6	123.4	14.4	99.3	97.3	27.30	6.26	630.0	95.8	604.0	10.07	718820	546016	172
2	18	60	30.08	324.6	141.6	156.4	19.4	128.1	131.9	23.60	7.16	662.0	95.3	630.9	10.51	753358	574308	179
3	18	60	30.04	314.0	168.3	183.0	18.9	152.7	152.7	26.74	9.52	813.5	95.1	773.6	12.89	935767	703744	222
4	18	60	30.03	311.6	185.4	200.1	18.7	168.1	171.4	24.54	9.71	849.5	95.0	807.0	13.45	966867	733967	232
5	19	60	30.09	322.0	121.6	136.4	19.3	102.4	100.3	48.04	11.36	1027.5	95.6	982.3	16.37	1170867	894089	276
6	19	60	30.05	321.0	144.6	159.4	19.0	129.1	134.6	47.20	14.51	1129.5	95.3	1076.4	17.94	1285652	979309	306
7	19	60	30.00	315.6	183.9	198.6	19.1	154.2	157.5	50.66	18.41	1423.0	95.3	1356.1	22.60	1624472	1234051	390
8	19	60	29.92	309.0	194.2	208.9	20.1	169.8	169.7	46.94	18.59	1377.5	95.0	1308.6	21.81	1568619	1192135	376
9	19	60	29.89	332.0	143.9	158.6	19.3	107.5	101.0	75.70	18.42	1377.0	95.4	1332.7	22.21	1591777	1213024	378
10	20	60	29.98	328.0	160.4	175.1	19.3	129.0	134.5	74.36	22.84	1486.5	95.5	1419.6	23.66	1697700	1292120	405
11	20	60	30.00	321.0	213.0	227.7	18.5	155.6	161.2	73.94	27.31	1836.0	96.8	1777.2	29.62	2132818	1616008	516
12	21	60	29.93	309.0	240.1	254.8	17.9	177.2	175.2	75.56	31.05	2279.5	97.6	2224.8	37.08	2673765	2021453	652
13	21	40	29.88	311.0	173.1	187.8	17.9	100.7	117.1	101.96	25.86	1913.3	96.7	1850.2	30.84	2214874	1681092	533
14	21	40	29.81	297.0	199.3	213.9	15.8	129.5	140.9	100.40	31.63	2185.5	97.5	2130.9	35.52	2555162	1930595	624
15	22	40	30.22	327.0	242.4	257.2	18.6	158.9	161.3	96.36	35.98	2493.8	97.1	2421.5	40.36	2910401	2202112	708
16	12	40	29.86	335.0	203.4	218.1	19.6	160.1	198.5	96.00	40.06	2824.5	98.4	2779.3	46.32	3345999	2530275	817
17	13	60	29.98	303.0	124.3	139.0	19.1	98.9	117.5	24.86	6.26	719.5	99.6	716.6	11.94	854259	652106	207
18	15	60	30.05	310.0	160.6	175.4	20.2	127.6	154.0	26.24	8.60	976.5	98.1	957.9	15.97	1145559	872743	278
19	15	60	30.00	300.0	188.3	203.0	19.7	147.9	184.2	24.88	9.61	1025.0	98.4	1008.6	16.81	1208605	918330	292
20	16	60	29.80	325.0	203.4	218.0	20.2	98.2	123.5	50.06	12.91	1110.0	99.9	1108.9	18.48	1323361	1009099	314
21	12	60	29.98	315.0	134.6	149.3	19.1	123.4	150.4	49.20	15.68	1322.5	99.3	1313.2	21.89	1571638	1196194	375
22	13	60	29.98	226.2	168.0	182.7	20.0	146.5	182.8	49.52	18.98	1541.0	98.7	1521.0	25.35	1823831	1385479	436
23	15	60	30.10	300.0	198.5	213.3	20.0	162.9	201.4	47.96	20.34	1651.0	98.3	1622.9	27.05	1950888	1477326	473
24	15	60	30.08	325.0	246.6	261.4	19.4	95.6	116.0	72.84	17.94	1268.0	97.9	1241.4	20.69	1482728	1130295	352
25	9	60	30.00	315.0	144.6	159.3	19.7	122.4	152.4	74.90	23.95	1697.5	98.4	1670.3	27.84	2001186	1517300	489
26	9	60	29.91	288.0	185.2	199.9	17.7	146.4	181.3	75.46	28.78	1963.0	98.2	1927.7	32.13	2314975	1750737	564
27	9	60	29.80	315.0	224.0	238.6	17.6	162.5	198.5	73.54	30.90	2137.8	98.8	2112.1	35.20	2538111	1918209	619
28	10	60	29.79	309.0	238.2	252.8	17.6	99.9	128.5	99.74	26.50	1872.0	98.7	1847.7	30.79	2212990	1679929	539
29	10	60	29.81	306.0	180.2	194.8	18.4	122.1	154.0	99.70	32.03	2309.3	99.0	2286.2	38.10	2745040	2076784	668
30	11	40	29.99	328.8	223.3	238.0	17.7	148.1	184.3	97.26	37.62	2573.3	97.9	2519.3	41.99	3029206	2288532	740
31	11	40	29.95	325.0	251.8	266.5	17.7	162.2	197.9	99.14	41.55	2666.0	98.1	2615.3	43.59	3147775	2371554	776
32	11	40	29.91	326.0	275.6	290.3	16.5											

Total B.T.U. rejected by pump and used in heating feed water, per hour.	Number of B.T.U. consumed by pump per hour.	DATA FROM PUMP END: Pressure in main on delivery side, pounds gage.	Pressure on suction side, inches of mercury.	Pressure corresponding to that given in Col. 21, pounds gage.	Vertical distance between gages, inches.	Total pressure of pump in pounds.	Temperature of suction water, Degrees Fahr.	Specific gravity of suction water.	Total discharge Head - feet.	Water delivered per hour in pounds.	Theoretical pump discharge from plunger displacement in lbs. per hr.	Slip, calculated. Pounds per hour.	Number of double strokes per minute.	Length of stroke in feet	Total feet of piston travel per hour.	Duty calculated from Col. 19, Foot - pounds.
18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34
546016	172804	197.6	5.5	2.7	18.4	195.8	49.7	1.003	450.3	24156	26289	2133	14.97	.912	1638	62946732
574308	179050	250.3	5.1	2.5	"	248.7	48.2	1.003	572.0	19914	22726	2812	12.65	.933	1416	63618431
703744	222023	299.5	5.1	2.5	"	297.9	48.0	1.003	685.2	22822	25743	2921	14.80	.903	1604	70409975
733967	232900	325.9	4.7	2.3	"	324.5	47.4	1.003	746.4	20712	23625	2913	13.52	.908	1472	66378003
594089	276518	198.8	2.0	1.0	"	198.7	47.9	1.003	457.0	44408	46254	1846	25.35	.948	2882	73392893
979309	306343	250.3	2.4	1.2	"	250.0	45.0	1.004	574.5	42997	45495	2498	24.60	.959	2832	80634375
234051	390421	299.5	2.2	1.1	"	299.3	45.9	1.004	687.7	46009	48837	2828	27.15	.933	3040	81041720
192135	376484	325.9	2.4	1.2	"	325.6	47.1	1.003	748.2	41348	45238	3890	24.27	.967	2816	82172347
213024	378753	198.8	1.8	.9	"	198.8	48.6	1.003	457.3	69861	72897	3036	37.45	1.017	4542	84348995
292120	405580	250.3	4.3	2.1	"	249.1	47.4	1.003	573.0	68734	71613	2879	34.95	1.064	4462	97106810
616008	516810	299.5	3.9	1.9	"	298.5	47.8	1.003	686.6	68207	71195	2988	35.68	1.036	4436	90615363
021453	652312	325.9	4.3	2.1	"	324.7	45.4	1.004	746.1	69412	72838	3426	36.97	1.022	4534	79391900
681092	533782	198.8	5.1	2.5	"	197.2	45.0	1.004	453.1	96720	98868	1548	48.85	1.044	6117	82100625
930595	624567	250.3	3.9	1.9	"	249.3	45.0	1.004	572.8	94440	96823	2383	48.68	1.031	6027	86612376
202112	706289	299.5	3.5	1.7	"	298.7	45.4	1.004	686.4	84775	92871	8096	46.15	1.044	5781	82155109
530275	815724	325.9	3.5	1.7	"	325.1	37.2	1.002	747.8	85496	92445	6949	46.30	1.037	5760	78376883
652106	202153	198.5	1.2	0.6	"	198.8	145.8	.978	468.9	18475	23349	4874	12.88	.965	1492	42853321
872743	272810	250.5	1.2	0.6	"	250.8	151.8	.975	593.4	17578	24556	6978	13.48	.973	1574	38234614
918330	290275	299.3	1.2	0.6	"	299.6	149.0	.976	708.2	17221	23315	6094	12.88	.966	1493	42015028
967953	306290	325.7	1.2	0.6	"	326.0	154.6	.972	773.7	17313	22395	5082	12.75	.941	1440	43733285
009099	314262	198.5	1.2	0.6	"	198.8	147.6	.977	469.4	42391	46958	4567	25.60	.978	3004	63317663
196194	375444	250.5	1.2	0.6	"	250.8	142.0	.981	589.8	41127	46240	5113	25.17	.977	2946	64606650
385479	438352	299.3	1.2	0.6	"	299.6	149.0	.976	708.2	40867	46411	5544	25.15	.985	2972	66024586
477326	473562	325.7	1.2	0.6	"	326.0	150.0	.976	770.6	38757	44944	6187	24.36	.984	2878	63067020
130295	352433	198.5	1.8	0.9	"	198.5	135.9	.984	465.4	65298	68803	3505	35.32	1.031	4370	85228274
517300	483886	250.5	1.2	0.6	"	250.8	142.4	.981	589.8	66717	70536	3819	37.05	1.011	4494	81320159
750737	564238	299.3	1.2	0.6	"	299.6	144.6	.979	706.0	65902	70931	5029	36.22	1.042	4528	82459551
918209	619902	325.7	1.2	0.6	"	326.0	146.1	.978	769.0	62293	69046	6753	34.87	1.055	4412	77275629
679929	533061	198.5	1.2	0.6	"	198.8	142.6	.980	468.0	82786	93800	11014	46.15	1.081	5982	77681828
076784	668256	250.5	1.2	0.6	"	250.8	155.2	.972	595.3	82757	93033	10276	47.10	1.058	5982	73722110
288532	740674	299.3	1.2	0.6	"	299.6	141.8	.981	704.6	81382	91584	10202	45.30	1.074	5835	77418348
371554	776221	325.7	1.2	0.6	"	326.0	142.0	.981	766.6	78496	93358	14862	44.33	1.118	5948	77523068

Table I.

STEAM END.							DATA FROM PUMP END												
Steam consumed per hour, in pounds.	Quality of steam, per cent.	Dry steam consumed per hour in pounds.	Dry steam consumed per minute in pounds.	Total B.T.U.supplied pump per hour.	Total B.T.U.rejected by pump and used in heating feed water, per hour.	Number of B.T.U.consumed by pump per hour.	Pressure in main on delivery side, pounds, gage.	Pressure on suction side, inches of mercury.	Pressure corresponding to that given in Col.21, pounds, gage.	Vertical distance between gages, inches.	Total pressure of pump in pounds.	Temperature of suction water, Degrees Fahr.	Specific gravity of suction water.	Total discharge Head - feet.	Water delivered per hour in pounds.	Theoretical pump discharge from plunger displacement in lbs. per hr.	Slip, calculated. Pounds per hour.	Number of double strokes per minute.	
13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	
630.0	95.8	604.0	10.07	718820	546016	172804	197.6	5.5	2.7	18.4	195.8	49.7	1.003	450.3	24156	26289	2133	14.97	
662.0	95.3	630.9	10.51	753358	574308	179050	250.3	5.1	2.5	"	248.7	48.2	1.003	572.0	19914	22726	2812	12.65	
813.5	95.1	773.6	12.89	925767	703744	222023	299.5	5.1	2.5	"	297.9	48.0	1.003	685.2	22822	25743	2921	14.80	
849.5	95.0	807.0	13.45	966857	733967	232900	325.9	4.7	2.3	"	324.5	47.4	1.003	746.4	20712	23625	2913	13.52	
1027.5	95.6	982.3	16.37	1170667	894089	276518	198.8	2.0	1.0	"	198.7	47.9	1.003	457.0	44408	46254	1846	25.35	
1129.5	95.3	1076.4	17.94	1285652	979309	306343	250.3	2.4	1.2	"	250.0	45.0	1.004	574.5	42997	45495	2498	24.60	
1423.0	95.3	1356.1	22.60	1624478	1234051	390421	299.5	2.2	1.1	"	299.3	45.9	1.004	687.7	46009	48837	2828	27.15	
1377.5	95.0	1308.6	21.81	1568619	1192135	376484	325.9	2.4	1.2	"	325.6	47.1	1.003	748.2	41348	45238	3890	24.27	
1377.0	95.4	1332.7	22.21	1591777	1213024	378753	198.8	1.8	.9	"	198.8	48.6	1.003	457.3	69861	72897	3036	37.45	1.
1486.5	95.5	1419.6	23.66	1697700	1292120	405580	250.3	4.3	2.1	"	249.1	47.4	1.003	573.0	68734	71613	2879	34.95	1.
1836.0	96.8	1777.2	29.62	2132818	1616008	516810	299.5	3.9	1.9	"	298.5	47.8	1.003	686.6	68207	71195	2988	35.68	1.
2279.5	97.6	2224.8	37.08	2673765	2021453	652312	325.9	4.3	2.1	"	324.7	45.4	1.004	746.1	69412	72838	3426	36.97	1.
1913.3	96.7	1850.2	30.84	2214874	1681092	533782	198.8	5.1	2.5	"	197.2	45.0	1.004	453.1	96720	98268	1548	48.85	1.
2185.5	97.5	2130.9	35.52	2555162	1930595	624567	250.3	3.9	1.9	"	249.3	45.0	1.004	572.8	94440	96823	2383	48.68	1.
2493.8	97.1	2421.5	40.36	2910401	2202112	708289	299.5	3.5	1.7	"	298.7	45.4	1.004	686.4	84775	92871	8096	46.15	1.
2824.5	98.4	2779.3	46.32	3345999	2530275	815724	325.9	3.5	1.7	"	325.1	37.2	1.002	747.8	85496	92445	6949	46.30	1.
719.5	99.6	716.6	11.94	854259	652106	202153	198.5	1.2	0.6	"	198.8	145.8	.978	468.9	18475	23349	4874	12.88	
976.5	98.1	957.9	15.97	1145559	872743	272810	250.5	1.2	0.6	"	250.8	151.8	.975	593.4	17578	24556	6978	13.48	
1025.0	98.4	1008.6	16.81	1208605	918330	290275	299.3	1.2	0.6	"	299.6	149.0	.976	708.2	17221	23315	6094	12.88	
1083.0	98.1	1062.4	17.71	1274243	967953	306290	325.7	1.2	0.6	"	326.0	154.6	.972	773.7	17313	22395	5082	12.75	
1110.0	99.9	1108.9	18.48	1323361	1009099	314262	198.5	1.2	0.6	"	198.8	147.6	.977	469.4	42391	46958	4567	25.60	
1322.5	99.3	1313.2	21.89	1571638	1196194	375444	250.5	1.2	0.6	"	250.8	142.0	.981	589.8	41127	46240	5113	25.17	
1541.0	98.7	1521.0	25.35	1823831	1385479	438352	299.3	1.2	0.6	"	299.6	149.0	.976	708.2	40867	46411	5544	25.15	
1651.0	98.3	1622.9	27.05	1950888	1477326	473562	325.7	1.2	0.6	"	326.0	150.0	.976	770.6	38757	44944	6187	24.38	
1268.0	97.9	1241.4	20.69	1482728	1130295	352433	198.5	1.8	0.9	"	198.5	135.9	.984	465.4	65298	68803	3505	35.32	1.
1697.5	98.4	1670.3	27.84	2001186	1517300	483886	250.5	1.2	0.6	"	250.8	142.4	.981	589.8	66717	70536	3819	37.05	1.
1963.0	98.2	1927.7	32.13	2314975	1750737	564238	299.3	1.2	0.6	"	299.6	144.6	.979	706.0	65902	70931	5029	36.22	1.
2137.8	98.8	2112.1	35.20	2538111	1918209	619902	325.7	1.2	0.6	"	326.0	146.1	.978	769.0	62293	69046	6753	31.87	1.
1872.0	98.7	1847.7	30.79	2212990	1679929	533061	198.5	1.2	0.6	"	198.8	142.6	.980	468.0	82786	93800	11014	46.15	1.
2309.3	99.0	2286.2	38.10	2745040	2076784	668256	250.5	1.2	0.6	"	250.8	155.2	.972	595.3	82757	93033	10276	47.10	1.
2573.3	97.9	2519.3	41.99	3029206	2288532	740674	299.3	1.2	0.6	"	299.6	141.8	.981	704.6	81382	91584	10202	45.30	1.
2666.0	98.1	2615.3	43.59	3147775	2371554	776221	325.7	1.2	0.6	"	326.0	142.0	.981	766.6	78496	93358	14862	44.33	1.

Table I.

Pounds per hour.	Number of double strokes per minute.	Length of stroke in feet	Total feet of piston travel per hour.	RESULTS. Duty calculated from Col.19. Foot - pounds.	Percentage of slip.	Theoretical capacity in gallons per hour.	Actual capacity in gallons per hour.	Overall mechanical efficiency, per cent.	Pounds of dry steam per I.H.P. per hour.	Pounds of dry steam per 1000 pounds of water pumped.	Total B.T.U. consumed by pump per hour, assuming feed to be heated to 200 degrees Fahr.	Duty calculated from Col.41. Foot - pounds.
30	31	32	33	34	35	36	37	38	39	40	41	42
33	14.97	.912	1638	62946732	8.11	3140.9	2886.2	91.7	96.49	25.00	621054	17514495
12	12.65	.933	1416	63618431	12.37	2715.3	2379.4	88.3	88.11	31.68	652666	17452737
21	14.80	.903	1604	70409975	11.35	3075.8	2726.7	89.0	81.26	33.90	803006	19467643
13	13.52	.908	1472	66378003	12.33	2822.6	2474.6	86.4	89.11	38.96	899221	18421175
46	25.35	.948	2682	73392893	3.99	5526.4	5305.9	90.2	86.47	22.12	1012601	20041908
98	24.60	.959	2832	80634375	5.49	5430.5	5132.4	86.0	74.18	25.03	1113913	22175679
28	27.15	.933	3040	81041720	5.79	5829.3	5491.8	86.8	73.66	29.48	1409197	22452779
90	24.27	.967	2816	82172347	8.60	5399.8	4935.4	84.0	70.39	31.65	1361521	22722069
36	37.45	1.017	4542	84348995	4.16	8709.6	8347.2	87.6	72.35	19.08	1378141	23181543
79	34.95	1.064	4462	97106810	4.02	8556.1	8212.2	87.0	62.21	20.65	1471189	26770851
88	35.68	1.036	4436	90615369	4.20	8506.2	8149.0	86.6	65.08	26.06	1846465	25362477
26	36.97	1.022	4534	79391900	4.70	8694.2	8285.6	84.3	71.65	32.05	2312097	22398841
48	48.85	1.044	6117	82100625	1.58	11729.7	11544.3	85.5	71.55	19.13	1915405	22879669
03	48.68	1.031	6027	86612376	2.46	11557.1	11272.8	86.4	67.37	22.57	2207583	24504280
96	46.15	1.044	5781	82155109	8.72	11085.4	10118.7	81.7	67.30	28.56	2518987	23100381
49	46.30	1.037	5760	78376889	7.52	11045.1	10214.5	80.6	69.38	32.51	2888899	22130891
74	12.88	.965	1492	42853321	20.88	2861.0	2263.6	69.8	114.47	38.79	734106	11800659
78	13.48	.973	1574	38234614	28.42	3018.2	2160.5	61.3	111.38	54.49	987925	10558276
94	12.88	.966	1493	42015028	26.14	2862.9	2114.6	64.1	104.95	58.57	1042015	11704162
82	12.75	.941	1440	43733285	22.69	2761.3	2134.8	66.8	104.88	61.36	1099678	12180900
67	25.60	.978	3004	63317663	9.73	5760.4	5199.9	77.8	85.89	26.16	1136973	17501150
13	25.17	.977	2946	64608850	11.06	5649.1	5024.3	78.1	83.75	31.93	1353182	17925678
44	25.15	.985	2972	66024586	11.95	5698.9	5017.9	76.8	80.14	37.22	1571820	18413056
87	24.38	.984	2878	63067020	13.77	5518.7	4758.8	74.1	79.79	41.87	1684680	17728081
505	35.32	1.031	4370	86228274	5.09	8379.7	7953.2	85.5	69.20	19.01	1278398	23771696
319	37.05	1.011	4494	81320159	5.41	8617.5	8151.3	83.0	69.74	25.04	1726527	22791238
039	36.22	1.042	4528	82459551	7.09	8682.7	8067.1	81.7	66.98	29.25	1998137	23285096
753	31.87	1.055	4412	77275629	9.78	8460.2	7632.8	78.3	68.35	33.91	2188038	21893275
014	46.15	1.081	5982	72681828	11.74	11470.8	10124.2	73.8	69.72	22.32	1907006	20216584
276	47.10	1.058	5982	73722110	11.05	11470.8	10203.3	77.7	71.38	27.63	2364954	20831374
202	45.30	1.074	5835	77418348	11.14	11188.9	9942.4	77.0	66.97	30.96	2618333	21900101
862	44.33	1.118	5948	77523068	15.92	11405.6	9589.8	73.1	62.94	33.32	2720920	22115694

Additional observations taken were boiler-pressure, suction and discharge temperatures. The barometer was read once during each test. All thermometers, gages, etc., were calibrated before and after each group of tests and corrections made when necessary.

RESULTS OF TESTS.

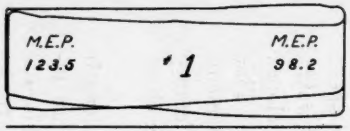
Table I gives the average of the data taken and the results as calculated, according to the appendix attached; Figs. 9 and 10 contain the curves of double strokes plotted successively against the discharge of the pump in pounds per hour, steam per I.H.P. hour, and the I.H.P. for discharge pressures of 200, 250, 300 and 325 pounds per square inch, gage, on the discharge side. Small numerals near individual points refer to tests as tabulated in Table I.

Figs. 1 to 8, show specimen cards from both ends of the pump at piston speeds of 50 and 100 feet per minute and extend over the entire range of discharge pressures. Cards 2 and 3 are at 50-foot piston speed with cold water, and cards 1 and 4 with hot feed water. Cards 5 and 8 at 100-foot piston speed are with cold water, and cards 6 and 7 are with hot feed water.

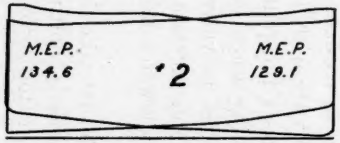
At the conclusion of the cold tests the pump developed several leaks, so that a general packing was necessary. This may have resulted in slight changes in valve adjustment, which would account for the noticeable increase in difference in top and bottom M.E.P. during hot-water tests over those in cold-water tests.

It was found that the pump would not take the full length of stroke smoothly at low piston speeds, halting for an instant near the end of the bottom to top stroke. This was especially noticeable at 25 feet per minute and necessitated one change of valve adjustment during the cold-water tests and again during the hot-water tests. All other necessary adjustments could be made with the cushion valves, as a slightly greater cushioning was required for quiet running at high than at low pressures. It is to be seen by reference to

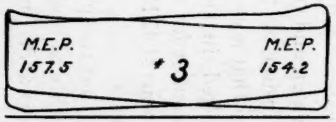
SCALE OF SPRING	100
LENGTH OF STROKE	0.978'
DISCHARGE PRESSURE	200
DOUBLE STROKES	25.60
TOTAL I.H.P.	12.91



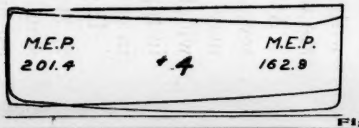
SCALE OF SPRING	100
LENGTH OF STROKE	0.959'
DISCHARGE PRESSURE	250
DOUBLE STROKES	24.60
TOTAL I.H.P.	14.51



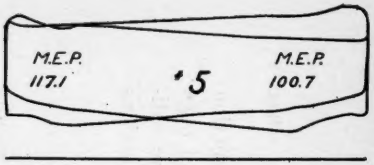
SCALE OF SPRING	150
LENGTH OF STROKE	0.933'
DISCHARGE PRESSURE	300
DOUBLE STROKES	27.15
TOTAL I.H.P.	18.41



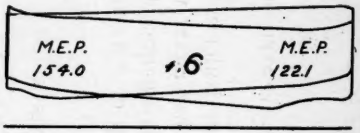
SCALE OF SPRING	150
LENGTH OF STROKE	0.984'
DISCHARGE PRESSURE	325
DOUBLE STROKES	24.38
TOTAL I.H.P.	20.34



SCALE OF SPRING	100
LENGTH OF STROKE	1.044'
DISCHARGE PRESSURE	200
DOUBLE STROKES	48.85
TOTAL I.H.P.	25.86

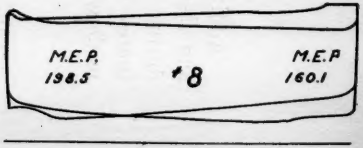


SCALE OF SPRING	150
LENGTH OF STROKE	1.058'
DISCHARGE PRESSURE	250
DOUBLE STROKES	47.10
TOTAL I.H.P.	32.03



SCALE OF SPRING	150
LENGTH OF STROKE	1.074'
DISCHARGE PRESSURE	300
DOUBLE STROKES	45.30
TOTAL I.H.P.	37.62

SCALE OF SPRING	150
LENGTH OF STPOKE	1.037'
DISCHARGE PRESSURE	325
DOUBLE STROKES	46.30
TOTAL I.H.P.	40.06



Figs. 1 to 8.

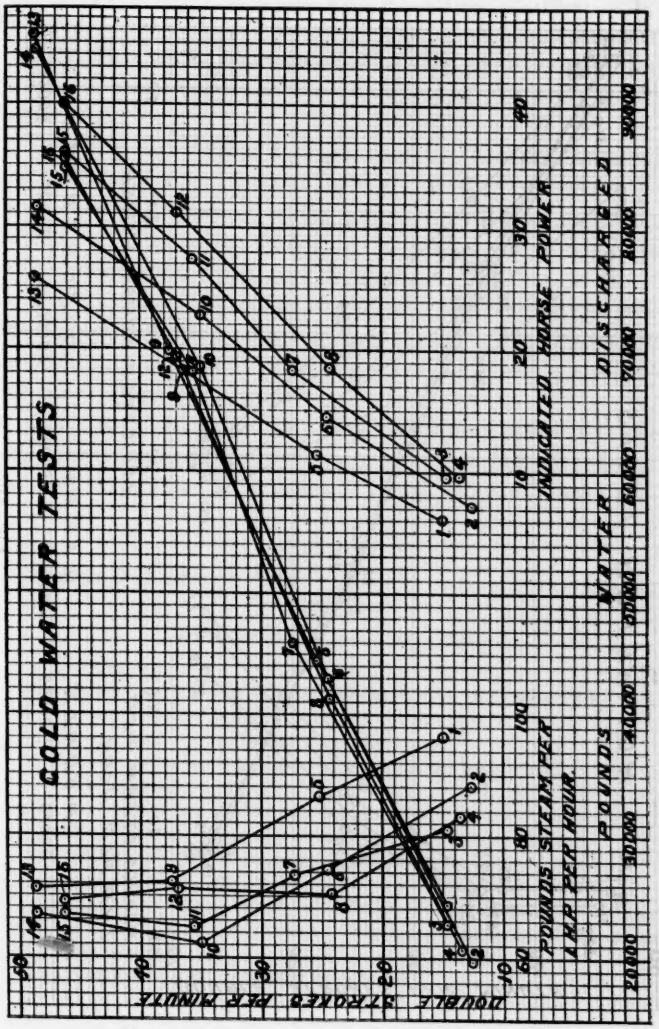

COLD WATER TESTS
DOUBLE STROKES PER MINUTE
POUNDS STEAM PER I.H.P. PER HOUR
INDICATED HORSE POWER
WATER DISCHARGED
POUNDS

Fig. 9.

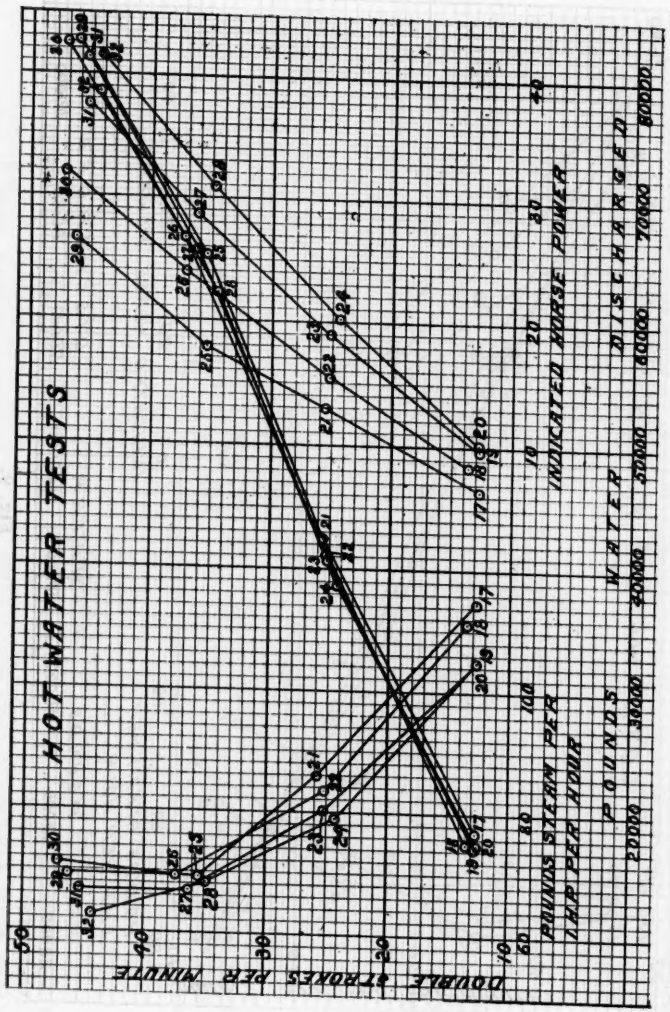

HOT WATER TESTS
DOUBLE STROKES PER MINUTE
POUNDS STEAM PER I.H.P. PER HOUR
INDICATED HORSE POWER
WATER DISCHARGED
POUNDS
10000
20000
30000
40000
50000
60000
70000
80000
10
20
30
40
50
60
70
80
90
100

Fig. 10.

column 32, length of stroke in feet, that the hot tests, Nos. 17 to 24 inclusive, were run with somewhat increased length of stroke over corresponding cold tests, Nos. 1 to 8 inclusive, indicating a better adjustment of moving parts and the possibility of closely approaching the rated stroke even at slow speeds.

The slip varied from 1.58 per cent. to 12.37 per cent., with an average of 6.59 per cent., when handling cold water, and from 5.09 per cent. to 28.42 per cent., with an average of 13.87 per cent., when handling hot water, values which are quite moderate for this type of pump.

Except for hot-water tests at 25-foot piston speed the steam consumption of the pump is satisfactory. The results show a minimum with most pressures when the piston speed is at or near 75 feet per minute.

Except for increased slip and consequent decreased overall mechanical efficiency the pump handled hot water equally as well as cold water, was easy to start and ran quietly.

APPENDIX I.

REPORT OF TEST OF 10-INCH × 7-INCH × 12-INCH BLAKE BOILER-FEED PUMP.

Method of obtaining results in Table I.

Items.

1, 2, 3. From data.

4. From data taken at beginning of each test and corrected by Smithsonian Meteorological Tables.

5, 6. From data average of five observations.

7. Item 6 + barometer pressure in pounds per square inch.

8. From data.

9, 10. Average of five sets of indicator cards.

11. From direct reading stroke meter.

12. $\frac{(\text{Item } 9 \times A_t \times \text{item } 10 \times A_b) \times \text{item } 11}{2 \times 33{,}000}$, where

A_t = area of top end of cylinder in square inches = 78.54,

A_b = area of bottom end of cylinder in square inches = 75.40.

13. As weighed.

14. Calculated from data from calorimeter;

$$X = \frac{H - q}{r},$$

H = heat content of one pound of steam corresponding to pressure and superheat in calorimeter,

q = heat of liquid corresponding to item 7,

r = heat of vaporization corresponding to item 7.

Items.

15. Item 13 × item 14.

16. $\frac{\text{Item } 15}{60}$.

17. Item 15 × total heat of dry steam corresponding to pressure in item 7.

18. Assuming that the quality of exhaust steam is 0.85, and the steam is used in a feed-water heater of 90 per cent. efficiency, then this column would be obtained as follows:

$$(.85\,r + q) \times \text{item } 15 \times 0.9;$$

where

r = heat of vaporization corresponding to item 8,

q = heat of the liquid corresponding to item 8.

19. Item 17 — item 18.

20, 21. From data.

22. Item 21 × .491.

23. As measured.

24. Item 20 + item 22 + $\frac{\text{item } 23 \times 62.5}{1{,}728}$.

25. From data.

26. From chemist's density determination.

27. $\frac{\text{Item } 24 \times 144}{\text{Item } 26 \times 62.42}$;

62.42 = density of pure water at 4 degrees C.

28. From tank or Venturi measurements.

29. Item 11 × 60 × $(A_t + A_b)$ × 62.42 × item 26;

A_t = area of top end of plunger in square feet,

A_b = area of bottom end of plunger in square feet.

30. Item 29 — item 28.

31. From stroke counter.

32. $\frac{\text{Item } 11}{\text{Item } 31}$.

33. Item 11 × 60.

34. Duty = $\frac{H \times W \times 1{,}000{,}000}{h}$, where

H = total head in feet = item 27,

W = weight of water pumped per hour = item 28,

h = total heat units consumed by pump per hour = item 19.

35. $\frac{\text{Item } 30}{\text{Item } 29}$.

36. $(A_t + A_b)$ × item 33 × 7.4805, where

A_t = area of top end of water plunger in square feet,

A_b = area of bottom end of water plunger in square feet.

37. $\frac{\text{Item } 36 \times (100 - \text{item } 35)}{100}$.

38. $\frac{\frac{H \times W}{60 \times 33{,}000}}{\text{I.H.P.}} \times 100$, where

H = total head in feet = item 27,

W = Weight of water pumped per hour = item 28,

I.H.P. = item 12.

Items.

39. $\frac{\text{Item 15}}{\text{Item 12}}$.

40. $\frac{\text{Item 15} \times 1{,}000}{\text{Item 28}}$.

41. Assuming that the condensed steam from the pump is available as boiler feed at 200 degrees F., then the total B.T.U.s absorbed by the pump per hour will be :

Item 13 $\times$ (item 14 $\times$ $r + q - q^1$), where

r = heat of vaporization corresponding to item 7,

q_1 = heat of the liquid corresponding to item 7,

q = heat of the liquid corresponding to 200 degrees F.

42. Duty $= \frac{H \times W \times 1{,}000{,}000}{h}$, where

H = total head in feet = item 27,

W = weight of water pumped per hour = item 28,

h = heat absorbed by pump per hour = item 41.

ADDRESS.

By Dean Mortimer L. Cooley, of the University of Michigan.

To the Graduating Class of 1912, School of Marine Engineering, United States Naval Academy.

I bring a message from inland to the seaboard, from the young men who are preparing themselves to engage in the upbuilding of our nation's life from the inside, to the young men who are preparing themselves to safeguard and protect our nation's life from the outside. Were we to dispense with the services of either there would be no need of the services of the other; for without the one there would be no growth and development, and therefore no country to protect; and without protection there would be no country to develop. Our interests are the same and we are mutually dependent, and while our work is along different lines—widely different in a technical sense—it is all work in the nation's vineyard.

It is not so much my purpose to dwell upon your future careers as engineers as to point out the work which can be done in a different line and which will greatly stimulate the growth and increase the strength of the fibers of our body politic. Our nation is not yet a homogeneous body, our framework is not yet well knit and we have not yet conditioned ourselves for the race of the nations about to begin. We are both lean and fat, and to him who is willing to see, there is a powerful tendency to the conditions making for fat. We flatter ourselves by saying we are great, and we do not see that our greatness is merely conceit.

The race of the nations, I said, is about to begin; it is already well under way; but only the lean-jawed have heard the start-

ing shot, yet they cannot spring from the mark, being held by a leash which cannot be loosed at will. There is the mental inertness, torpitude, of the people to be dealt with before the leash can be loosed. We hear that the Navy has the backing of the people. True, but for the most part it is a sentimental backing. Our people are proud of the Navy—vastly proud—but for the reason mainly that it contributes pleasure; it affords opportunity for hero worship to the sentimental, and boasting to the conceited. With danger in the horizon which can be sensed, action is speedy. All of the great advances in our American Navy have been the result of some conspicuous cause apparent to everyone. What the Navy needs is a comprehension at all times of its importance.

How can such a comprehension be brought about? I am here to tell you how it can at least be aided now, and eventually brought about. Educate the people to look upon the Navy as a pure business proposition, devoid of all sentiment, in the same way that they look upon their own business. Have them understand that the Navy is as great a necessity to their own business as other protection for which they are constantly demanding legislation and getting it. Have them understand that the Navy is an insurance, and a protection in the same sense, only infinitely more so, as the insurance which every man takes out on his own property. Such a comprehension of the Navy needs a campaign of education. It probably will need more than that—another war, to bring the lesson home. The lesson has been brought home before, but it does not stay home—that is the need for a campaign of education.

Those who have had an opportunity to see the navies of other countries and who have an inkling of the great game being played will easily be convinced, but they are relatively few. Not all can come to the Navy; therefore, the Navy should go to them. It was a wise law, in the interests of the Navy, which for a few years following 1879 provided for the detail of officers to institutions of learning. It was the initial step away from the seaboard towards a dry-land Navy. A dry-land Navy, let me say to you, is fully as important as a

sea-going Navy. The time is easily within the recollection of some of the younger men present when more than three-quarters of our sailors were from the coast, and foreign born at that, and they were an old lot; but now where do our sailors come from? How many of them are foreign born, and how old are they? They—these sailors from the prairies—got their first idea from the dry-land Navy, and once they demonstrated their worth the sea-going Navy got an idea also, and thus has come about in large measure the present Navy.

It would be a splendid thing to revive that old law and scatter throughout this broad land of ours representative officers to duty in connection with our institutions of learning. The possibilities of good are scarcely measureable. There would, first of all, be the influence on the vast number of young men and women attending our universities and colleges. They would come into personal contact with the Navy and come to know it at first hand. Then there would be the opportunity to tell these young men and women in the formative period of their careers what the Navy is really for, and what might happen without it. They are a wonderful lot, these young men and women, in the higher schools of learning; they are eager for information, and they would make good use of it. They are at an age when only the truth will satisfy. In a few years some of them will be in Congress, where they are very much needed. Their sons and daughters will grow up with a better understanding of our nation's need, and the Monroe Doctrine will no longer be classed in history with the Dred Scott case, but will be the live issue it really should be.

There would also be great good brought about through the association of officers detailed for such duty with the faculties. Their acquaintance would become more or less intimate, and opportunity would be afforded for the safe disclosure of information now withheld from the general public. I cannot too strongly emphasize the good which might come about in that way. It would pave the way towards that comprehension of the real purpose of the Navy which is so much needed.

Such are a few of the things which the people need and which the naval officer can give. In return the naval officer can receive benefit. In the first place he will be taken out of his rather narrow field of work and come into contact with the great human forces which he feels but does not understand. He would come to know the great changes which are constantly being made in the engineering schools, changes in the scope of work and the opportunities for specialization. He would come into contact with specialists in a great variety of subjects, not alone in engineering, but in history, philosophy, political economy and law. He would become acquainted with some of the great problems being worked out in civil life and which are coming on to the stage in swift succession. All this would be good for the Navy.

But there are other good things to be accomplished, and a reference to some of them may perhaps be more in accord with the spirit of this occasion. There are coming to the fore with tremendously swift strides such subjects as production engineering and efficiency engineering. The problem is to produce economically and to operate with high efficiencies. These are not mere mechanical questions, but embrace the human force, both physical and mental. One of the great questions is to limit costs and provide against depreciations of various kinds. The world is just waking up to the fact that the greatest factor in depreciation is not due to wear and exposure to the elements but to obsolescence. The wonderful advances in many of the arts require the scrapping of machines long before they are worn out.

Nowhere is this true more than in the Navy. Not only are vast sums expended for battleships, but their useful life has been growing shorter and shorter. It is difficult for the layman to understand why this should be so; and no doubt it is one of the reasons tending to curtail appropriations. Can it be met? In some measure, yes. If the useful life of a battleship be only five or ten years, and they must be consigned to the scrap-heap before being half worn out, why not spend less money on them? Why not recognize at the outset that their life will be short and build accordingly?

The Government has the reputation of building always in the best manner possible regardless of expense, and contractors know that and bid accordingly. Thus in the Navy a ship may cost 10 cents a pound or more as against half that or less in civil life, and its useful life be no longer, nor, indeed, anywhere near as long.

It is not so much my idea to criticise the Navy in this matter of high costs as to suggest that by a proper contact of the officers of the Navy with work as it is done outside of the Navy, the Department would be in a position to know of itself whether such economies were possible. The civilian's criticism of Navy costs may be entirely wrong, but he does not know it, nor does the officer, unless he knows how the civilian builds.

I have been asked to criticise this School from which you are today being graduated. I will be frank in what I have to say. The idea of the school is a splendid one and it has great possibilities. I have been greatly interested in its success. It is, of course, an experiment, and even if finally it shall not have given in full measure the results anticipated it will have been well worth the trial. Oftentimes a negative result is just as useful as a positive result, as it may point out the right course even more distinctly than the more favorable result.

This School is essentially a research school. It suggests the fundamental idea of a university as defined by an admirer of Mark Hopkins, which was a log in the woods with Mark Hopkins sitting on one end and the student on the other. There can be no true teacher without true students. The teacher merely points the way, the student pursues it. Each of you has come to the age of original thought, the time when you accept theories or statements only when you have yourself found them to be true. You are supposed to have judgment. Your method of work is the same as that followed by young men who are training themselves to become efficient teachers in our universities, and the object of work, while it may seem to be different, is not really different, for your aim is to

become specialists, authorities, in different lines, and it is expected of you to take back into the service the results of your labors; thus you will become teachers.

The success of the research idea depends wholly on the student. For one student it is ideal, for another a failure. It need not take long for the sincere student to know himself in this respect. Once he has found what nature has done for him, his aim should be to aid nature in his development along that line. The prime requisite for a specialist is aptitude.

I have questioned whether a greater measure of good might not come out of a plan which would scatter you for a time, a half year say, among the big universities where you could live with a specialist in some line and become imbued with his spirit. This you could best do by taking part with him in his investigations. Then later you could come together here at Annapolis and continue your work, each becoming a teacher to the others.

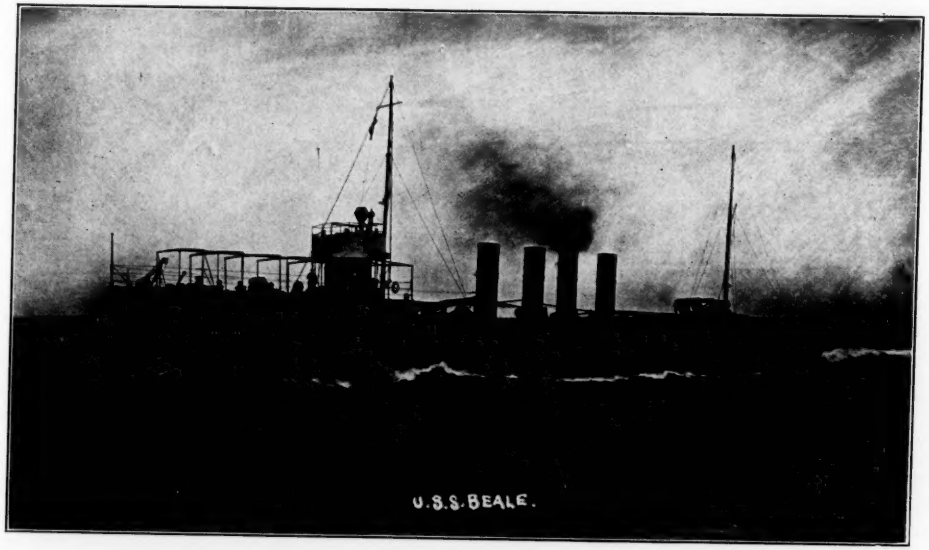
I think the idea of living for a time with some of our big manufacturing concerns should not be abandoned; but your status should be a different one. You should not be guests to be entertained, but engineers in the same sense as other young men of your own age and experience whom you find there from the engineering schools. It should be the aim of these concerns to see that you become acquainted with their ways of doing business for profit; not how they get business, but how it is done once it is gotten.

And finally, I would urge the importance of giving something in return for what you get, that being a knowledge of the Navy which will command its support for purely business reasons. Thus you will be most fruitful, as the seed you can sow will produce and return many fold.

In conclusion, I would impress on you the fact, as disclosed to me by my own experience, that nowhere can one find an education doing more for one than right here in Annapolis. But little teaching is ever done; in my own day I heard but two lectures. It could hardly be said that we were taught at all—we simply had to learn everything ourselves. Such a

system seems brutal to my colleagues in university life, where too much of the learning, like a bad tasting medicine, is put, as it were, into capsules to be swallowed without effort. Here it was more like taking quinine off a knife blade, but the result was magnificent in that it developed self-reliance, and produced men of stamp and character, such as was exemplified in the man who carried a message to Garcia.

Young gentlemen, you were courageous to seek assignment to this post-graduate school. You have taken upon yourselves a great responsibility. Much will be expected of you. See that you make good.



U.S.S. BEALE.

U. S. S. *BEALE.*

DESCRIPTION AND TRIAL PERFORMANCE.

By Henderson B. Gregory, Associate.

Torpedo-Boat Destroyer No. 40, the *Beale*, is one of the six vessels authorized by an Act of Congress approved June 24, 1910. She is a triple-screw Parsons turbine vessel designed for a speed of 29½ knots, at a trial displacement of 742 tons and with the main turbines developing 12,000 shaft horsepower. She was built under contract by the William Cramp and Sons' Ship and Engine Building Co., of Philadelphia, Pa. The contract was signed December 1, 1910, the price being $654,000, and the time of construction twenty-four months.

PRINCIPAL HULL DIMENSIONS.

Length on L.W.L., feet and inches	289–00
overall, feet and inches	293–10½
Breadth, extreme, at L.W.L., feet and inches	26–01½
Ratio of length to beam	11.08
Draught to L.W.L., feet and inches	8–04
Displacement corresponding, tons	742
per inch at L.W.L., tons	12
Area immersed midship section, square feet	144.5
Co-efficient of fineness, block	0.408

GENERAL DESCRIPTION OF HULL.

The hull is of steel, galvanized in all parts below the water line.

The vessel has a raised forecastle, which is carried aft to frame No. 41. On the forecastle deck are located the anchors, capstan and pilot house, with bridge above. Above the bridge is a searchlight platform on which is mounted one searchlight.

The main deck is a weather deck aft of frame No. 41. On this deck and under the forecastle are located a small store room, lamp room and windlass-engine room, wardroom officers' quarters, pantry and ship's galleys. In a deck house aft are located the crew's water closets and washroom, and wireless room. There are two masts fitted with signal yards and wireless outfit.

The berth deck is in two parts, extending from the stem to frame No. 51 and from frame No. 124 aft, the space between being interrupted by the deep fuel-oil tanks, cofferdam, engine and firerooms. On this deck forward are located store rooms, paints and oils room and crew's quarters, and aft are located crew's and petty officers' quarters and store rooms.

In the hold, from forward aft, are the forward trimming tanks, chain lockers, store rooms, fresh-water tank, magazines and handling rooms, fuel-oil tanks, cofferdam (extending to main deck and arranged to be used as a reserve-feed tank), boiler and engine rooms, fuel-oil tanks, magazine and handling room and the after trimming tanks. There is a reserve-feed tank, built in the ship's structure, in the forward end of the engine room, and similarly constructed fuel-oil settling tanks extending along the sides of the firerooms outboard of the boilers. There is also an engineer's store room in the port forward corner of the forward fireroom.

BATTERY.

The battery consists of five 3-inch semi-automatic guns, one on the forecastle, commanding ahead and broadside fire, two on the main deck forward, port and starboard, at break of forecastle, one on main-deck center line abaft of engine-room hatch, and the other on the main-deck center line well aft.

There are also two 0.30-caliber automatic guns mounted on the forecastle, port and starboard.

The torpedo outfit consists of three 5.2-m. by 45-cm. twin torpedo tubes mounted on the main deck, one aft on the center line and the other two, port and starboard, amidship.

There is a four-stage Ingersoll-Rand air compressor for the

torpedoes, located in the engine room. Its capacity is 20 cubic feet of free air at 2,500 pounds per square inch per minute. The steam cylinder is 8 inches in diameter by 5-inch stroke, and the four compressor cylinders are $9\frac{1}{4}$, $4\frac{3}{4}$, $3\frac{1}{8}$ and $1\frac{1}{4}$ inches in diameter, respectively. No accumulator is fitted.

BOATS CARRIED.

The following small boats are carried in davits at the vessel's side:

1 20-foot power dory;
2 20-foot whale boats.
1 14-foot wherry.

ANCHOR WINDLASS.

The anchor windlass is located on the main deck at frame No. 12. It is of the vertical type, manufactured by the American Ship Windlass Co. The engine is of the following principal dimensions:

Number of cylinders	2
Diameter of cylinders, each, inches	4
Stroke, inches	6

STEERING ENGINE AND GEAR.

There are three steering stations: (1) The pilot house fitted with combined steam and hand gear; (2) top of pilot house, steam gear only; (3) the hand gear on the main deck aft. The steering engine is located in the pilot house and is of the following principal dimensions:

Type	Williamson Bros.
Number of cylinders	2
Diameter of cylinders, inches	5
Stroke, inches	5

FIRE MAIN.

The fire main extends throughout the machinery space on the port side close under the main-deck beams. It is $2\frac{1}{2}$ inches in diameter, which size is carried forward to frame No.

38 and aft to frame No. 132 in the crew's spaces. The main is supplied by the three fire and bilge pumps in the engine and firerooms, the pump connections being the full size of the main. Branches for the various fire plugs are taken off the main at convenient locations. All fire plugs are 1½ inches in diameter and located as follows:

Plug No. 1, crew's quarters, berth deck, frame No. 28, port;

Plug No. 2, main deck, frame No. 47, center line;

Plug No. 3, main deck, frame No. 82, port side of smoke-pipe No. 3;

Plug No. 4, main deck, frame No. 104, port side of engine hatch;

Plug No. 5, main deck, frame No. 130, port side of deck house;

Plug No. 6, forecastle deck, frame No. 32, center line.

In addition to the regular fire plugs above enumerated there is a 1½-inch hose valve on the discharge manifold of each fire and bilge pump.

Forward and aft there are 2-inch connections from the fire main for magazine flooding and sprinkling.

A 1¼-inch connection, dividing into three 1-inch branches, is taken off the fire main aft for flushing out the stern-tube bearings.

FLUSHING SYSTEM.

The flushing system is taken off the fire main direct. There is a 2-inch connection forward supplying the officers' bath, galleys and pantry, and one of same size aft for the crew's water closets and wash room.

FRESH-WATER SYSTEM.

Fresh water is carried in the ship's tank, located in the forward hold, and having a capacity of about 3,250 gallons. The tank has two 2½-inch filling connections, port and starboard respectively, fitted with hose valves at the ship's sides. There is also a 1-inch filling connection from the distilling apparatus.

Located on the forecastle deck, immediately abaft the pilot house, is a small gravity tank of 50-gallons capacity, for use of the galleys. The tank is fitted with steam coil to prevent freezing in cold weather. A hand pump is provided on the main deck at frame No. 42 for filling the gravity tank.

There are no fresh-water connections to any of the lavatories.

DRAINAGE SYSTEM.

A 3½-inch main drain is led throughout the machinery space on the port side, and connected to all fire and bilge pumps by suctions the full size of the main. Macomb strainers are fitted in these suctions close to the pumps. There is a 3½-inch bilge suction from the main in each fireroom and two of the same size in the engine room, fitted with plate strainers at the bilge ends.

For draining the compartments abaft the engine room, there is a 2½-inch branch reducing to 2 inches as it leads aft, from which 1½-inch suctions are led to the various compartments requiring drainage and the trimming tanks.

An independent drainage system is provided for the forward hold, consisting of a small steam-driven bilge pump (see Table I), located on the berth deck at frame No. 28. This pump has suctions from all the compartments requiring drainage and the trimming tanks.

VENTILATION.

All ventilation is by natural means, except in the boiler rooms which are ventilated by the forced draft blowers.

HEATING SYSTEM.

The usual heating system, with brass-pipe coil radiators, is installed. Steam for the quarters forward is taken off the auxiliary steam line in the forward boiler room, and that for the after compartments from the auxiliary steam line in the engine room.

TABLE I.—PUMPS AND PUMP CONNECTIONS.

No.	Pumps.	Size, inches.	Kind.	Inches.	Suction pipes from—	Inches.	Discharge pipes to—	Location.
2	Main air.....	11 × 2 22-inch × 16.	Twin, vert., bucket, single acting.	10	Condensers through water seal.	8	Feed tank.	In engine room.
				4	Condensers over water seal.			
				1	Reserve feed tanks.			
2	Main circulating.	28 in. diam. of runner.	Vert., single engine; centrifugal pumps.	16½	Sea	16	Main condenser.	In engine room.
				7	Bilge.....	3	Oil cooler.*	
2	Main feed.....	14½×10×12	Vert., piston, double acting, single.	4½	Feed suction pipe and reserve feed tanks.	4	Main feed disch.	In engine room.
				4½	Channel ways.			
2	Aux. feed.....	14½×10×12	Vert., piston, double acting, single.	4½	Feed suction pipe and reserve feed tanks.	3½	Auxiliary feed.	One in each boiler room.
				1½	Hose connection.	1½	Hose connection.	
				2½	For'd reserve feed tank.			
2	Fireroom fire and bilge.	7×7×12	Vert., piston, double acting, single.	3½	Sea.....	2½	Fire main.	One in each boiler room.
				3½	Drainage	3	Overboard.	
				1½	Hose connection.	1½	Hose connection.	
1	Engine-room fire and bilge.	7×7×12	Vert., piston, double acting, single.	4	Sea	2½	Fire main.	In engine room.
				3½	Drainage.....	3½	Overboard.	
						2	Distiller.	
						2	Flushing system through fire main.	
				1½	Hose con'tion	1½	Hose con.	
1	Auxiliary air and circulating.	6×8×8×7	Combined, hor., piston, double acting, single.	4	Aux. condenser.	3	Feed tanks.	In engine room.
				4	Sea	4	Aux. condenser.	
1	Evap. feed....	3½×4×4	Vert., piston, double acting, single.	1½	Sea	1	Evaporators.	In engine room near evaporators.
				1½	Dist. circulat'g water disch.			
1	Oil-cooler circulating.	7×7×12	Vert., piston, double acting, single.	2½	Sea	2½	Oil cooler ..	In engine room.
2	Lubricating oil.	4½×5×6	Vert., piston, double acting. single.	2½	Lubricating oil tank.	2½	Through cooler to bearings.	In engine room.
4	Fuel-oil service.	5×3×6	Vert., piston, double acting, duplex.	2	Settling tanks,	1½	Burners	Two in each boiler compartment.
2	Fuel-oil tank supply.	7×7×12	Vert., piston, double acting, single.	4½	Storage tanks.	4	Set. tanks..	One in each boiler compartment.
				4½	Deck hose connection.	4	Stor. tanks..	
1	Bilge.....	3½×4×4	Vert., piston, double acting, single, light service.	2	Forward compartments.	1½	Overboard..	Under forecastle, location to be determined by C. & R.
				1½	Hose connection.			

*One pump only.

MAIN ENGINES.

The vessel is equipped with Parsons turbines, designed to run at 775 revolutions per minute, when developing 12,000 S.H.P. The turbines are arranged on three lines of shafting as shown in Figure 1.

The arrangement provides five ahead and two astern turbines, each L.P. turbine including an astern turbine in the after end of its casing.

For ahead motion the center shaft is driven by the M.H.P. turbine, and the outboard shafts by the L.P. ahead turbines alone or in combination with the H.P. cruising and I.P. cruising turbines, as described in the following paragraphs:

Full Speed Ahead.—But three turbines are used for this purpose, steam being admitted to the M.H.P. turbine and expanded through the L.P. ahead turbines into the condensers. Under this condition the astern and cruising turbines revolve idly in a vacuum, which is maintained through the drain connections.

High Cruising Speeds.—Four turbines are used at these speeds, steam being admitted to the I.P.C. turbine, thence through the M.H.P. turbine and L.P. ahead turbines, exhausting into the condensers; the remaining turbines revolving idly in a vacuum.

Low Cruising Speeds.—All five ahead turbines are used for these speeds, steam being admitted to the H.P.C., I.P.C., M.H.P. and L.P. ahead turbines, in the order named, and exhausted into the condensers; the astern turbines revolving idly in a vacuum.

For astern motion the outboard shafts alone are used. The center shaft does not reverse.

Non-return valves are fitted in the receiver pipes between all turbines as shown in Figure 1, to prevent back flow of steam when changing turbine combinations or reversing.

The turbines are controlled from the working platform, the arrangement of operating valves and piping being shown in Figure 2.

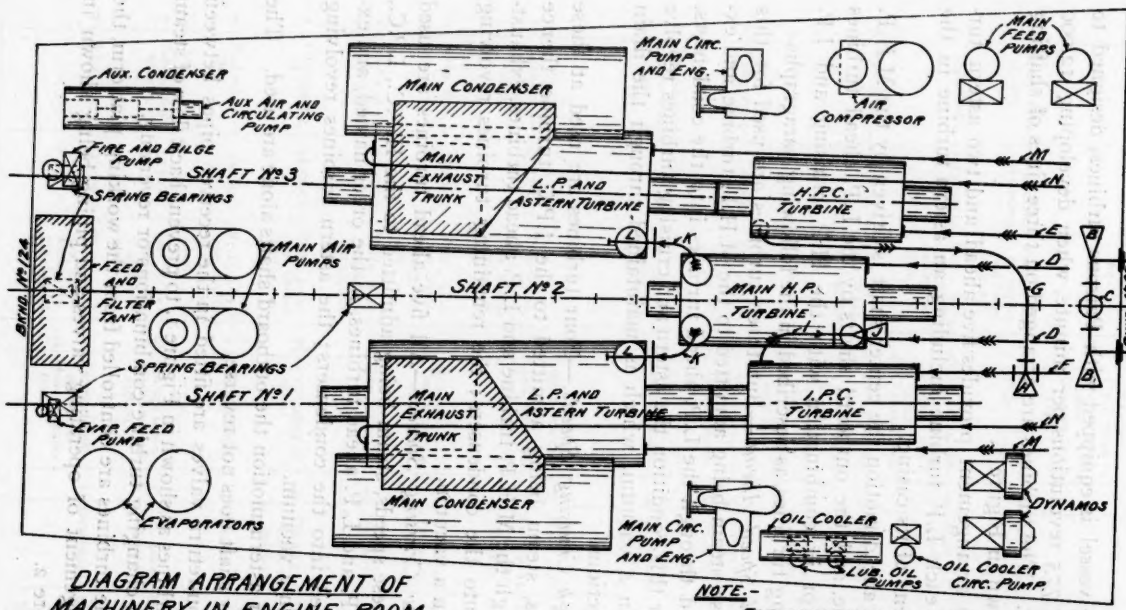
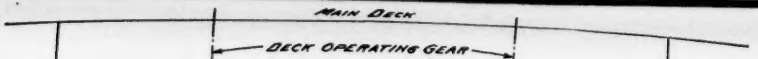
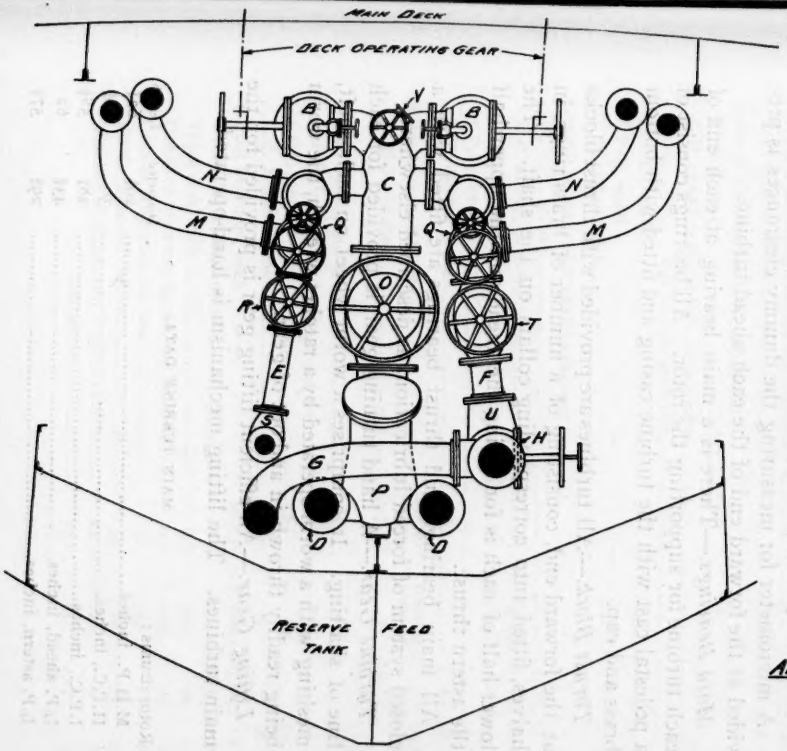


DIAGRAM ARRANGEMENT OF MACHINERY IN ENGINE ROOM U.S.S. BEALE

FIG. 1.

NOTE:- FOR ARRANGEMENT OF VALVES AND PIPES AT WORKING PLATFORM AND KEY TO PIPES AND VALVES SHOWN HEREON, SEE FIG. 2.





SECTION AT FR. #99
LOOKING FOR'D.

MARK	DESCRIPTION
A	9" MAIN STEAM FROM BOILERS
B	9" " " STOP VALVE
C	12" COMBINED STEAM
D	9" STEAM TO M.H.P. TURBINE
E	5" " " H.P.C. "
F	8" " " I.P.C. "
G	9" " FROM H.P.C. TO I.P.C. TURBINES
H	9" NON-RETURN CROSS VALVE
I	13" STEAM FROM I.P.C. TO M.H.P. TURBINES
J	13" NON-RETURN ANGLE VALVE
K	16" STEAM FROM M.H.P. TO L.P. AND. TURBINES
L	16" NON-RETURN ANGLE VALVE
M	6" MANEUVERING STEAM TO L.P. AND. TURB.
N	6" " " " " L.P. AST. "
O	12" THROTTLE VALVE, M.H.P. TURBINE
P	12" STEAM STRAINER
Q	6" MANEUVERING VALVE
R	5" THROTTLE VALVE, H.P.C. TURBINE
S	5" STEAM STRAINER
T	8" THROTTLE VALVE, I.P.C. TURBINE
U	8" STEAM STRAINER
V	2½" BLEEDER VALVE

U. S. S. BEALE.

ARRANGEMENT OF VALVES AND PIPING
AT
WORKING PLATFORM
U.S.S. BEALE

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FIG. 2.

Turbines.—Each turbine is composed of two essential parts; first, the fixed part or cylinder, and second, the moving part or rotor. In the cylinder are mounted the guide blades and on the rotor the moving blades. The cylinders are of hard, close-grained cast iron, divided into two parts at the axis on a horizontal plane, the lower half being provided with feet for bolting to the seating. Each rotor is built up of a drum of forged steel, securely fastened to a cast-steel wheel at each end, that is shrunk on and keyed to the rotor shaft. The rotors are perfectly balanced when entirely completed, in order to prevent vibration in running.

The dummies and rotor-shaft glands are steam packed with the usual labyrinth packing.

A micrometer for measuring the dummy clearances is provided at the forward end of the each ahead turbine.

Main Bearings.—There is a main bearing at each end of each turbine for supporting the rotor. All bearings consist of a pedestal cast with the turbine casing and fitted with bottom brass and cap.

Thrust Block.—All turbines are provided with thrust blocks at the forward end, consisting of a number of brass rings, in halves, fitted into corresponding collars on the shaft. The lower half of each is for taking the ahead and the upper half the astern thrust.

All main bearings and thrust bearings are fitted with a closed system of forced lubrication, as described elsewhere.

Turbine Gear.—A hand turning gear is provided for each line of shafting. It comprises a worm wheel on each shaft, meshing with a worm operated by a ratchet wrench, the worm being readily thrown in and out of gear.

Lifting Gear.—An efficient lifting gear is provided for the main turbines. The lifting mechanism is hand-operated.

MAIN TURBINE DATA.

Rotor drums:	Diameter.	Length.
M.H.P., inches	29¼	62⅞
H.P.C., inches	30	50¼
I.P.C., inches	28¼	58½
L.P. ahead, inches	43¼	62
L.P. astern, inches	29¼	57½

Number of expansions:

M.H.P.	4
L.P. ahead	8
H.P.C.	3
I.P.C.	3
L.P. astern	4

Turbine casing, diameter, each expansion:

M.H.P., inches	32¼, 34, 35½, 38
L.P. ahead, inches	46¼, 48, 49½, 52, 54, 59, 59, 59
H.P.C., inches	30⅝, 31, 31⅛
I.P.C., inches	30⅝, 31½, 32
L.P. astern, inches	31½, 32½, 35, 35

Length of casing for each expansion and diameter noted above:

M.H.P., inches	13½, 14⅞, 17, 18½
L.P. ahead, inches	7, 7⅛, 7⅜, 8½, 8, 8¼, 8, 8
H.P.C., inches	16½, 17½, 17⅜
I.P.C., inches	18 11/16, 19⅝, 20 1/16
L.P. astern, inches	9¾, 21½, 14⅜, 15

Rows of blading for each expansion:

M.H.P., each	10
L.P. ahead, for first four expansions	5
last four expansions	4
H.P.C., each	18
I.P.C., each	16
L.P. astern, each	8

Length of blades for each expansion:

M.H.P., inches	1½, 2½, 2⅞, 4⅛
L.P. ahead, inches	1½, 2½, 2⅞, 4⅛, 5⅛, 7⅛, 7⅜, 7⅛
H.P.C., inches	⅝, ¼, 1 1/16
I.P.C., inches	1, 1½, 1⅜
L.P. astern, inches	⅝, 1⅜, 2⅜, 2⅜

Rotor shaft and bearings:	Length of bearing.	Diameter at bearing.	Diameter of axial hole.
M.H.P., inches	7	7	(for'd) 3 (aft) 3½
L.P. ahead, inches	12	7	3
H.P.C, inches	6½	6½	3½
I.P.C., inches	6½	6½	3½
L.P. astern, inches	12	7	3½

Thrust bearings, each:	M.H.P.	L.P. ahead.	H.P.C.	I.P.C.
Collars on shaft, number	13	13	7	7
Thickness, inch	0 9/16	0 9/16	0½	0½
Distance between, inch	0 1 1/16	0 1 1/16	0⅝	0⅝
Outside diameter, inches	9⅜	9⅜	8⅜	8⅜
Inside diameter, inches	5¼	5¼	5¼	5¼
Number of shoes, top	12	12	6	6
bottom	13	13	7	7

SHAFTING.

There are three lines of shafting, as shown in Figure 1. The wing shafts are in four sections each, consisting of two line shafts supported by two spring bearings, one stern-tube shaft supported by the two stern-tube bearings, and one propeller shaft supported by the strut bearing. The center shaft is in five sections, there being four line shafts supported by five spring bearings, and one stern-tube and propeller shaft supported by the stern-tube bearings.

All stern-tube and strut bearings are lined with lignum vitæ and the shafts are composition bushed at these bearings. The shafting within the stern tubes is covered with a composition casing.

The inboard coupling consists of a collar secured to the stern-tube shaft by four keys. Forward of this collar are two thin half collars fitted into a groove turned in the end of the shaft, to prevent the shaft from backing out, the whole being through bolted to the coupling disk on the line shaft.

The outboard coupling is of the ordinary sleeve type secured to the shafts by four feathers and two cross-keys each.

SHAFT DATA.

Line shaft, diameter outside, inches	6¼
axial hole, inches	3¼
Stern-tube shaft, diameter outside, inches	6¼
axial hole, inches	3¼
Propeller shaft, diameter outside, inches	6¼
axial hole, inches	3¼
Couplings, diameter, inches	12
thickness, inches	1½
Inboard coupling, diameter of collar, outside, inches	12½
inside, inches	6¼
length of collar, inches	5½
thickness of half collars, inch	0¾
Coupling bolts, number each coupling	8
diameter (taper) * at face of coupling, inches	1½
Outboard coupling, length of sleeve, inches	20¾
diameter of sleeve, outside, inches	9¼
Spring bearings, diameter, inches	6¼
length, inches	13½

* Parallel bolts in inboard couplings.

Forward stern-tube bearings, diameter, inches	7⅞
length, inches	21
After stern-tube bearings, diameter, inches	7⅞
length, inches	33
Strut bearings, diameter, inches	7⅞
length, inches	33

PROPELLERS.

There are three three-bladed propellers. The port propeller is left-hand and the center and starboard propellers are right-hand. The blades and hubs are of manganese bronze and cast in one piece. The blades are true screw machined to pitch.

PROPELLER DATA.

Diameter, feet and inches	5–03
Pitch, feet and inches	4–10
Ratio of pitch to diameter	0.921
Area, projected, square feet	12.83
helicoidal, square feet	14.44
disc, square feet	21.65
Ratio projected to disc	0.592
helicoidal to disc	0.667

MAIN CONDENSING APPARATUS.

Main Condenser.—There are two main condensers of elliptic-triangular section located outboard of the low-pressure turbines. They are of the bent-tube type, with the tubes expanded into the tube sheets. The principal dimensions are as follows:

Maximum width, inside, feet and inches	3–11½
depth, inside, feet and inches	7–07⅜
Length between tube sheets, feet and inches	11–09
Thickness of tube sheets, inches	01½
Tubes, number	2,494
diameter, inches	00⅝
thickness, B.W.G.	18
Cooling surface, square feet	4,800.7
Exhaust nozzle, area through, square feet	13.77
Diameter of air-pump dry suction, inches	07
wet suction, inches	06½
circulating-water inlet and outlet, inches	16

*Main Air Pumps.**—A Blake wet and dry, vertical, twin, bucket, single-acting air pump is provided for each condenser. Each pump has a single steam cylinder 11 inches in diameter, and two water cylinders 22 inches in diameter each, with a common stroke of 16 inches. The arrangement of the air-pump piping is shown in Figure 3.

*Main Circulating Pump and Engine.**—There is one centrifugal circulating pump for each main condenser, driven by a small reciprocating engine. The principal dimensions of the pump and engine are as follows:

Diameter of suction nozzle, inches	16½
discharge nozzle, inches	16
impeller, inches	28
engine cylinder, inches	7
Stroke, inches	6

Feed and Filter Tank.—A feed and filter tank of 715 gallons capacity is located at the after end of the engine room. The filter chamber is in the top of the tank, and has a capacity of 166 gallons. The filter is so arranged that the entering water will flow over the top of a plate and leave in like manner, insuring that the filter chambers are always filled with water and the filtering material submerged.

ENGINE-ROOM AUXILIARIES.

*Auxiliary Condenser, Air and Circulating Pump.**—There is an auxiliary condenser in the engine room of 256 square feet of cooling surface. It is connected through the auxiliary exhaust pipe to all the machinery. The condenser has a 6-inch × 8-inch × 8-inch × 7-inch Blake horizontal, simplex, double-acting, combined air and circulating pump.

*Main Feed Pumps.**—Two Blake 14½-inch × 10-inch × 12-inch main feed pumps of the vertical, double-acting, single type, are located in the engine room. The pumps have suctions from the main feed tanks and discharge to the boilers through the feed-water heaters or by-passes.

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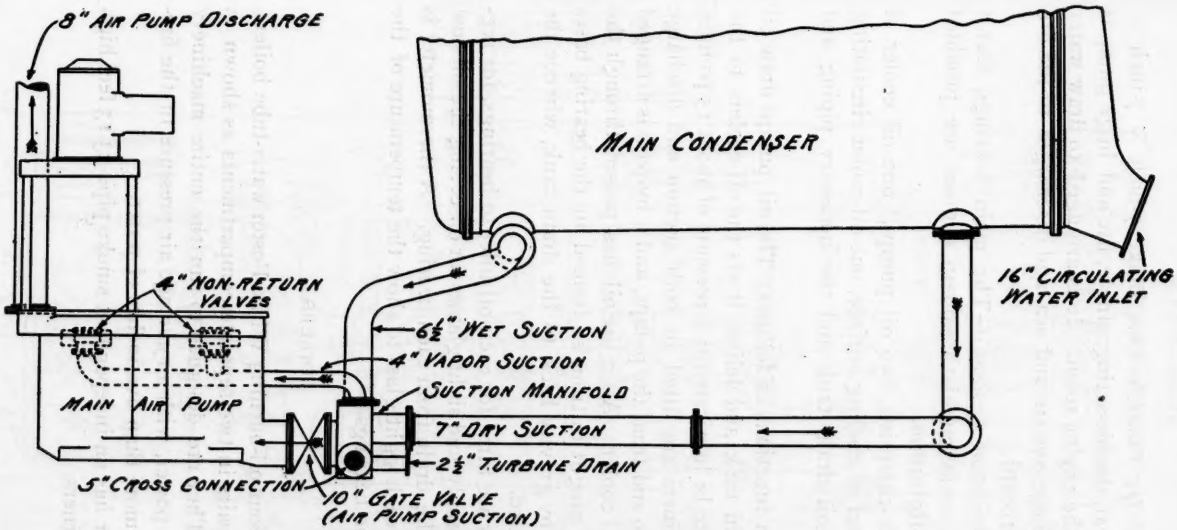
*See Table I.



ARRANGEMENT OF MAIN AIR PUMP SUCTION AND DISCHARGE PIPES.
U.S.S. BEALE.
FIG. 3.

*Fire and Bilge Pump.**—One Blake 7-inch × 7-inch × 12-inch vertical, double-acting, single, fire and bilge pump is provided in the engine room. It is arranged to draw water from the drainage system and sea, and discharges to the fire main and overboard.

Forced-Lubrication System. — The main bearings, thrust bearings and main circulating-pump engines are provided with forced lubrication.

The system comprises two oil pumps,* one oil cooler of 17.5 square feet of cooling surface, one oil-cooler circulating pump,* one oil drain tank and the necessary piping and fittings.

The system functions as follows: The oil pumps draw oil from the drain tank and deliver it via the oil coolers to the various parts to be lubricated at a pressure of about 15 pounds. Macomb strainers are fitted in both suction and discharge connections to and from the pumps, and a by-pass is arranged around the oil cooler. After the oil has passed through the bearings it is caught in troughs formed in the bearing bases and drained by gravity back to the drain tank, whence the cycle is repeated.

Lock cocks are fitted for each oil inlet to bearings, for regulating the supply, and sight glasses for observing the oil flow are fitted in the drain from each bearing. A thermometer is also fitted at each sight glass to show the temperature of the oil leaving the bearings.

BOILERS.

There are four oil-burning White-Foster water-tube boilers, arranged in pairs in two separate compartments as shown in Figure 4. They are designed to run the entire machinery plant at full power, with an average air pressure in the fire-rooms of not more than five inches of water.

Each boiler has an independent smoke pipe 33.13 feet high above the burners.

* See Table I.

MARK	DESCRIPTION
A	6½" BOILER STOP VALVE
B	6½" MAIN STEAM PIPE
C	9" " " "
D	9" EXPANSION JOINT
E	4" TRIPLE SAFETY VALVE

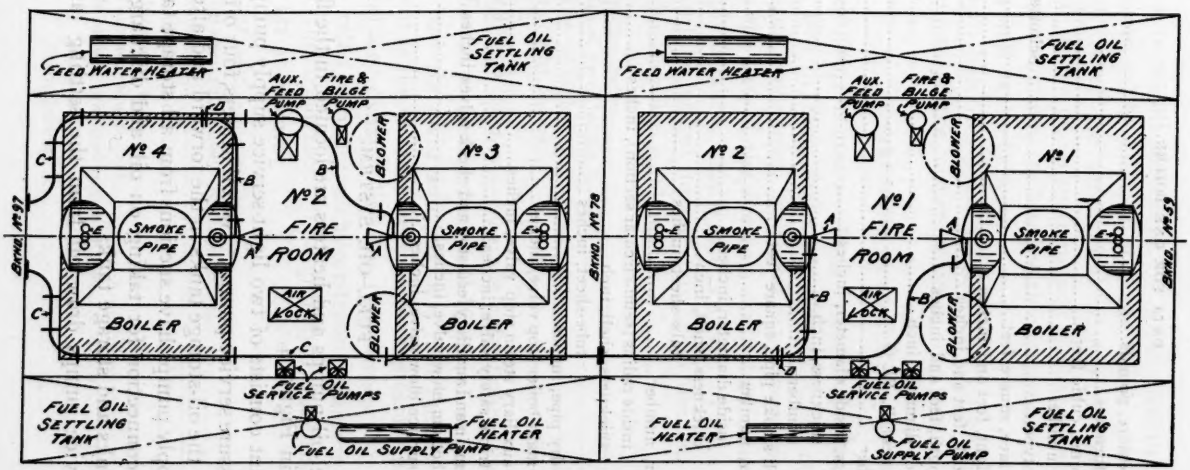


DIAGRAM
ARR'G'T OF MACHINERY IN FIRE ROOM
U. S. S. BEALE

FIG. 4.

DATA FOR ONE BOILER.

Working pressure, pounds	260
Number of furnaces	1
Furnace volume, cubic feet	400
Oil burners, number	9
type	Schutte-Koerting.
Heating surface, square feet	4,500
External height, feet and inches	12–$7\frac{9}{16}$
width, feet and inches	14–$7\frac{3}{8}$
length, feet and inches	11–$0\frac{3}{8}$
Tubes, outside diameter, inch	1
thickness, B. W. G	13
number	2,728
Down-comers, inside diameter, inches	$8\frac{1}{8}$
thickness, inch	$0\frac{3}{8}$
number	2
Area through smoke pipe, square feet	13.32
Drum, upper, number	1
inside diameter, inches	48
thickness, shell, inch	$0\frac{9}{16}$
tube-sheet, inches	$1\frac{9}{16}$
lower, number	2
inside radius (semi-circular section), inches	15
thickness, shell, inch	$0\frac{7}{16}$
tube-sheet, inches	$1\frac{9}{16}$
Diameter of dry pipe, inches	$6\frac{1}{4}$
main boiler stop valve, inches	$6\frac{1}{2}$
auxiliary steam stop valve, inches	3
triple safety valve, inches	4
main and auxiliary feed-stop and check valves, inches	$2\frac{1}{4}$
bottom-blow valve, inches	$1\frac{1}{4}$
surface-blow valve, inches	$1\frac{1}{2}$

FUEL–OIL SYSTEM.

The fuel-oil pumps and heaters are located in the firerooms as shown in Figure 4.

The plant consists of two light-service supply pumps,* four heavy-pressure service pumps,* two oil heaters, four oil-settling tanks and the oil-storage tanks in the forward and after holds.

The supply pumps have suctions from all storage tanks and the deck connection for taking on oil, and discharge to all settling tanks and storage tanks.

The service pumps draw oil from the settling tanks and

* See Table I.

deliver it to the oil burners on the boilers, via the oil heaters or by-passes. The burners, nine per boiler, of the Schutte-Koerting type, are mechanical atomizers. Strainers are provided in the pump suctions and discharges, and the supply pipes to burners are fitted with cocks controlled from the deck as well as the fireroom.

A small hand oil pump is installed in each fireroom for supplying oil to the burners when raising steam.

FIREROOM AUXILIARIES.

Forced-Draft Blowers.—Four forced-draft blowers are installed, two in each fireroom. The fans are of the B. F. Sturtevant single-inlet, horizontal type, mounted on vertical shaft in the base of the fireroom ventilators, from which the air supply is taken. Each fan is driven by a 24-inch Terry steam turbine located immediately below the fan.

The fan data are as follows:

Fan, diameter, inches	39
width at tip of blades, inches	7⅞
number of blades	8
Diameter of inlet, inches	30½

Feed-Water Heater.—There is a feed-water heater of the direct-flow type installed in each fireroom, on the discharge side of both the main and auxiliary feed pumps. The heating agent is the exhaust steam, a back pressure being kept in the auxiliary exhaust line for this purpose by means of a spring relief valve at each condenser connection, opening toward the condenser. The principal dimensions of each heater are as follows:

Diameter of shell, inside, inches	16
Length between tube sheets, feet and inches	6-04½
Tubes, number	240
diameter, inch	00⅝
thickness, inch	0.049
Heating surface, square feet	250.3
Diameter of feed inlet and outlet, inches	03½
auxiliary exhaust nozzle, inches	06
drain, inches	01¼

*Auxiliary Feed Pumps.**—There are two Blake, 14½-inch × 10-inch × 12-inch auxiliary feed pumps, one in each fireroom. They are of the vertical, double-acting, single type and are arranged to feed any boiler.

*Fire and Bilge pumps.**—In each fireroom there is a Blake 7-inch × 7-inch × 12-inch, vertical, double-acting, single fire and bilge pump. Each pump draws water from the drainage system and the sea, and discharges to the fire mains and overboard.

MAIN STEAM PIPING.

There are two main steam lines. The port line is connected to the two after boilers only, and that on the starboard side to the two forward boilers. The branches from the boilers are 6½ inches in diameter each, and the lines proper are 9 inches, uniting in the engine room into a common 12½-inch casting at the working platform. The general arrangement of the main steam piping is shown in Figs. 1, 2 and 4.

AUXILIARY STEAM PIPING.

There is a 3-inch stop valve on each boiler for supplying the auxiliary steam line, which leads throughout the machinery space with connections to the various auxiliaries and a branch leading forward to the deck machinery, forward heating system and galleys.

AUXILIARY EXHAUST PIPING.

An auxiliary exhaust pipe extends throughout the machinery space and elsewhere as required for the various auxiliaries. Connections are provided for direct exhaust into either main or auxiliary condensers, either feed-water heater, or into the atmosphere through the after escape pipe, at will. There are also connections for admitting exhaust steam into the L.P. ahead turbine's steam belt when desired.

* See Table I.

MAIN AND AUXILIARY FEED PIPING.

The main and auxiliary feed pumps take their suctions from the feed suction main, which extends from the bottom of the main feed tank to the auxiliary feed pump in the forward fireroom. The main is 7 inches in diameter in the engine room, decreasing in size to 6 inches in the after fireroom and 4½ inches in the forward fireroom. The suction branches to the main feed pumps are 5½ inches in diameter each and those for the auxiliary feed pumps 4½ inches. All feed pumps are arranged to feed any boilers via the main feed line and through the feed heater or by-passes. In addition to this provision, each auxiliary feed pump has direct connections to the boilers in its fireroom.

INTERIOR COMMUNICATION.

The customary engine and fireroom telegraphs, gongs, telephones, voice tubes, etc., are fitted for transmitting orders and signaling to the machinery compartments and other parts of the vessel.

EVAPORATING AND DISTILLING APPARATUS.

The distilling apparatus is installed in the after starboard corner of the engine room, as shown in Figure 1, with the distillers at high elevation in the after engine-room hatch above the main deck.

There are two evaporators and two distillers, with their accessories, arranged to operate in single effect. The plant has a combined capacity of 3,000 gallons of water per twenty-four hours.

DATA FOR ONE EVAPORATOR.

Type	Vertical, Reilly multicoil.
Diameter, inside, inches	32
Height overall, feet and inches	6–00
Number of tube coils	10
Tubes in coils, diameter, inches	01
thickness, B.W.G	16
Heating surface, square feet	61.5
Diameter of steam nozzle, inches	01¼
vapor nozzle, inches	02½
feed valve, inch	01
blow valve, inches	01¼

DATA FOR ONE DISTILLER.

Type	Modified Bu. of S. E.
Diameter, inside, inches	09
Length between tube sheets, feet and inches	2–05½
Tubes, number	43
diameter, inch	00⅝
thickness, B.W.G.	18
Cooling surface, square feet	17
Diameter of circulating-water inlet and outlet, inches	01¼
vapor inlet, inches	02
drain, inch	00½

MACHINE TOOLS.

For performing minor repairs one small Hendey-Norton, motor-driven lathe, of 12 inches swing and 6-foot bed, is installed together with all the necessary tools and attachments.

ELECTRIC PLANT.

The dynamos are located in the forward starboard corner of the engine room (see Figure 1). The installation consists of two horizontal, compound-wound, direct-current, 5-kilowatt Diehl generators, each driven by a Terry steam turbine. Each generator will deliver at normal load 40 ampères of current at 125 volts, when running 5,000 revolutions per minute.

TORSIONMETERS.

Each line of shafting is fitted with a Gary-Cummings torsionmeter for ascertaining the shaft horsepower of the main turbines.

TRIALS.

The contract required the following trials:

(*a*) A progressive trial over a measured-mile course for standardizing the screws, extending from maximum speed down to a speed of 12 knots.

(*b*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed obtainable, the average for the four hours not to be less than 29½ knots.

(*c*) An endurance and fuel-oil and water-consumption trial

TABLE II.—STANDARDIZATION TRIAL DATA U. S. S. "BEALE."

Course off Delaware Breakwater. June 25, 1912.

Number of run	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
Time on course, minutes.... seconds....	4 32.5	5 48.8	4 19.8	4 16.9	3 21.2	3 37.1	2 55.4	3 37.8	2 30.8	2 54.3	2 31.1	2 34.7	2 17.4	2 35.4	2 3 3	2 18.4	2 4.1	2 6.1	1 55.7	2 4.6	1 49.6	1 55.0	1 49.7	1 53.3	1 48.3
Revolutions, to go 1 knot:																									
Shaft No. 1	1,310.8	1,636.3	1,252.8	1,649.9	1,313.8	1,658.1	1,353.6	1,667.8	1,461.4	1,698.4	1,471.0	1,709.5	1,540.9	1,732.6	1,474.7	1,640.0	1,491.9	1,733.1	1,605.4	1,715.4	1,692.6	1,795.5	1,713 7	1,801.5	1,756.4
No. 2	1,154.2	1,440.5	1,104.7	1,427.2	1,139.2	1,393.2	1,144.0	1,406.1	1,244.8	1,442.9	1,265.7	1,457.4	1,307.4	1,467.3	1,411.0	1,570.7	1,418.4	1,589.4	1,467.7	1,570.9	1,541 7	1,613.6	1,550.6	1,598.5	1,543.0
No. 3	1,304.9	1,628.3	1,246.4	1,622.8	1,288.2	1,646.7	1,338.0	1,652.0	1,272.6	1,479.8	1,294.5	1,495.2	1,348.7	1,520.2	1,432.0	1,606.1	1,451.3	1,678 4	1,542.4	1,662.2	1,636.3	1,755.0	1,663.6	1,761.1	1,709.0
Average	1,256.6	1,568.4	1,201.3	1,566.6	1,247.1	1,566.0	1,278.5	1,575.3	1,326.3	1,540.4	1,343.7	1,554.0	1,399.0	1,573.4	1,439.2	1,605.6	1,453.9	1,666.9	1,538.5	1,649 4	1,623.5	1,721.4	1,642.7	1,720.4	1,669 5
Revolutions, per minute:																									
Shaft No. 1	288 7	281.7	284.4	385.3	391.9	458.2	462.9	459.3	581.6	584.7	584.5	663.2	672.9	669 0	717.3	710.7	721.3	824.6	832.9	826.4	926.6	936.8	936.9	954.4	974.0
No. 2	254.2	248.0	255.2	333.3	339.8	385.0	391.2	387.3	495.4	496.7	502.9	565.4	570.9	566.5	686.3	680.7	685.8	756.3	761.5	756.7	844.0	841.9	847.7	846.9	855.6
No. 3	287.4	280.3	290.0	378.9	384.3	455.0	457.6	455.0	506.5	509.4	514.4	580.1	589.0	586.9	696.5	696.0	701.7	798.6	800.2	800.7	895.8	915.6	909.5	933.0	947.7
Average	276.7	270.0	277.5	365.8	372.0	432.7	437.2	433.9	527.9	530.2	533.9	602.9	610.9	607.5	700.1	695.8	702.9	793.2	798.2	794.6	888.8	898.1	898.0	911.4	925.8
Speed in knots	13.21	10.32	13.86	14.01	17.89	16.58	20.53	16.53	23.87	20.65	23.83	23.27	26.20	23.17	29.19	26.01	29.01	28.55	31.12	28.89	32.85	31.30	32.82	31.77	33.24
S.H.P., Shaft No. 1	179	167	169	458	487	816	837	831	1,900	1,957	1,894	2,847	2,943	2,890	3,060	2,987	2,860	3,985	4,239	3,815	4,426	4,831	5,008	4,990	5,417
No. 2	124	114	125	108	110	426	433	439	1,030	1,072	1,072	1,572	1,588	1,591	3,095	3,069	3,110	4,104	4,256	4,046	6,813	5,024	5,127	5,076	5,105
No. 3	194	189	194	440	446	786	791	786	916	935	958	1,441	1,511	1,537	2,802	2,925	2,785	3,860	4,105	3,697	4,233	4,722	4,862	4,887	5,271
Total, all shafts	497	470	488	1,006	1,043	2,028	2,061	2,056	3,846	3,964	3,924	5,860	6,042	6,018	8,957	8,981	8,755	11,949	12,600	11,588	15,472	14,577	14,997	14,953	15,793

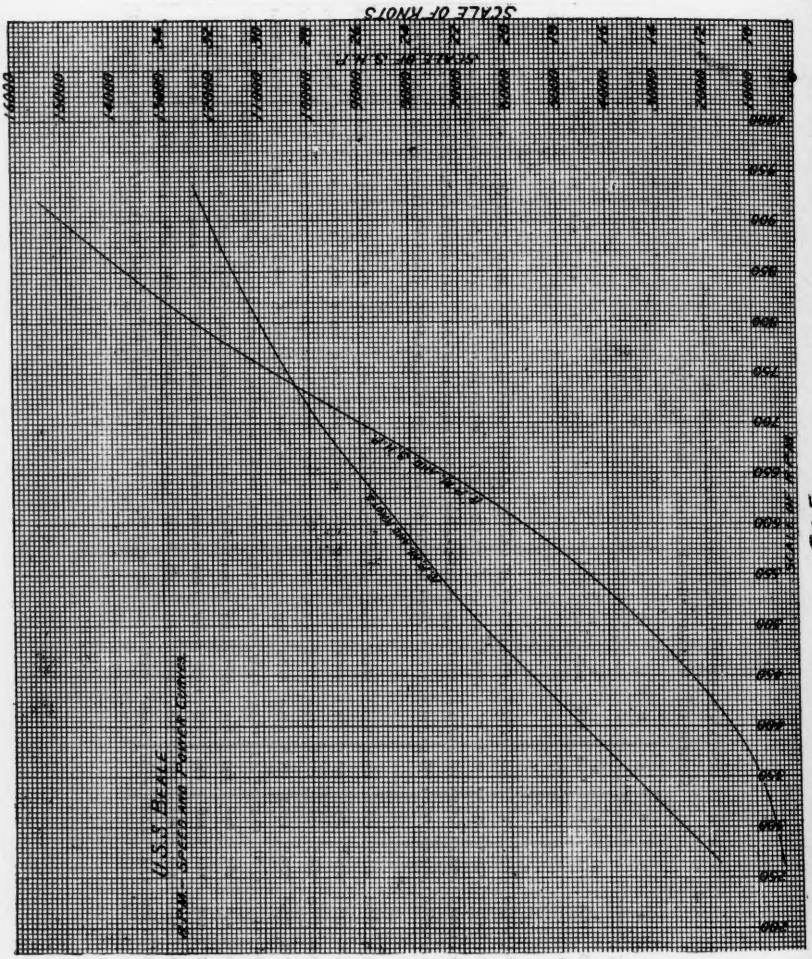


FIG. 5.

of four hours duration in the open sea in deep water, at as nearly as possible a uniform speed of 25 knots, the average not to fall below that figure. The trial to be conducted as nearly as possible to cruising conditions.

(*d*) An endurance and fuel-oil and water-consumption trial at 16 knots under conditions similiar to the preceding trial.

TABLE III.—U. S. S. "BEALE."—TRIAL DATA.

.....	4-hour full-power trial.	4-hour 25-knot trial.	4-hour 16-knot trial.
Date of trial	June 26, 1912.	June 26, 1912.	June 25, 1912.
Speed in knots	29.649	25	16.345
Slip of propellers, per cent. of own speed, mean :			
Starboard propeller	23.00	22.97	14.56
Middle propeller	17.22	8.45	— 0.51
Port propeller	21.47	12.40	14.37
Draught, mean on trial, feet and inches	8–2½	8–4¼	8–6$\frac{23}{32}$
Displacement corresponding, tons	740	761.5	774.6
Number of turbines used	3	4	5
boilers used	4	4	2
oil burners used (9 per boiler)	36	16	4
Heating surface used, square feet	18,000	18,000	9,000
Pressures (average) :			
Main steam, at boilers, pounds gage	256	255	257.8
steam chest, pounds gage	200 (M.H.P.)	242.5 (I.P.C.)	180 (H.P.C.)
Vacuum in condensers, inches of mercury	27.7	28.5	28
Air pressure in fireroom, inches of water	4.6	2.1	1.7
Revolutions per minute of main turbines (average) :			
Shaft No, 1	807.36	680.49	401.07
No. 2	750.90	572.56	340.95
No. 3	791.60	598.37	400.34
Mean, all shafts	783.28	617.14	380.79
Collective S.H.P., all shafts, from curve	11,800	6,250	1,345

The various trials enumerated above were run at Delaware Breakwater, on June 25 and 26, 1912. Owing to foggy weather the 16-knot endurance trial was run first, which was done the morning of the 25th, off the breakwater. At completion of the trial and anchor test that followed, the *Beale* steamed inside the Breakwater and anchored until the fog lifted.

The fog lifted in the early afternoon and the *Beale* got under way for the standardization runs over the measured mile. This trial began at 2:20 P. M. and was successfully completed at 6:55 P. M. the same afternoon. In all twenty-five runs were made over the measured mile, and the data obtained are given in Table II, from which the curves, Fig. 5, were plotted. From the data obtained it was found to require 776.5 r.p.m. of the main turbines to attain the contract speed of 29.5 knots, and 372 and 617 r.p.m. for 16 and 25 knots, respectively.

At the conclusion of the standardization runs the steering-gear tests and reversing and backing tests were made, after which the *Beale* anchored inside the Breakwater for the night.

On the following morning, June 26, the *Beale* steamed out into deep water and began the 25-knot endurance trial. The weather was hazy and the sea calm, with moderately long swell. The trial was concluded at 11:20 A. M. and immediately preparations were made for the four-hour, full-speed trial, which began at 12:30 P. M. the same day and terminated at 4:30 P. M. The contract speed was comfortably exceeded without special effort, and the data obtained on this and the two endurance trials are given in Table III.

U. S. S. *JOUETT* AND *JENKINS.*

CONTRACT TRIALS PERFORMANCE.

By Henderson B. Gregory, Associate.

The *Jouett* and the *Jenkins* (Torpedo-Boat Destroyers Nos. 41 and 42) are two of the six vessels authorized by an Act of Congress approved June 24, 1910. They were built under contract by the Bath Iron Works, of Bath, Maine, the price being $654,500.00 each, and the time of construction twenty-four months from date of contract, November 30, 1910.

They are triple-screw Parsons turbine vessels, designed to develop 12,000 S.H.P. at about 775 revolutions per minute of the turbines, the contract speed being 30 knots and the displacement about 742 tons.

The vessels are in all principal features similar to the *Paulding* and *Drayton*, built by the same contractors two years previous, a full description of which appeared in the November, 1910, number of the Journal; hence, it will not be repeated here.

TRIALS.

The following trials were required of both vessels:

(*a*) A progressive trial over a measured-mile course for standardizing the screws.

(*b*) A full-speed trial of four hours' duration in the open sea at the highest speed obtainable. The average speed to be at least 30 knots, with the average air pressure in the fire-rooms not exceeding 5 inches of water and the steam pressure at the M.H.P. turbine not to exceed 240 pounds above the atmosphere.

(*c*) An endurance and fuel-oil and water-consumption trial of four hours' duration in the open sea, at as nearly as pos-

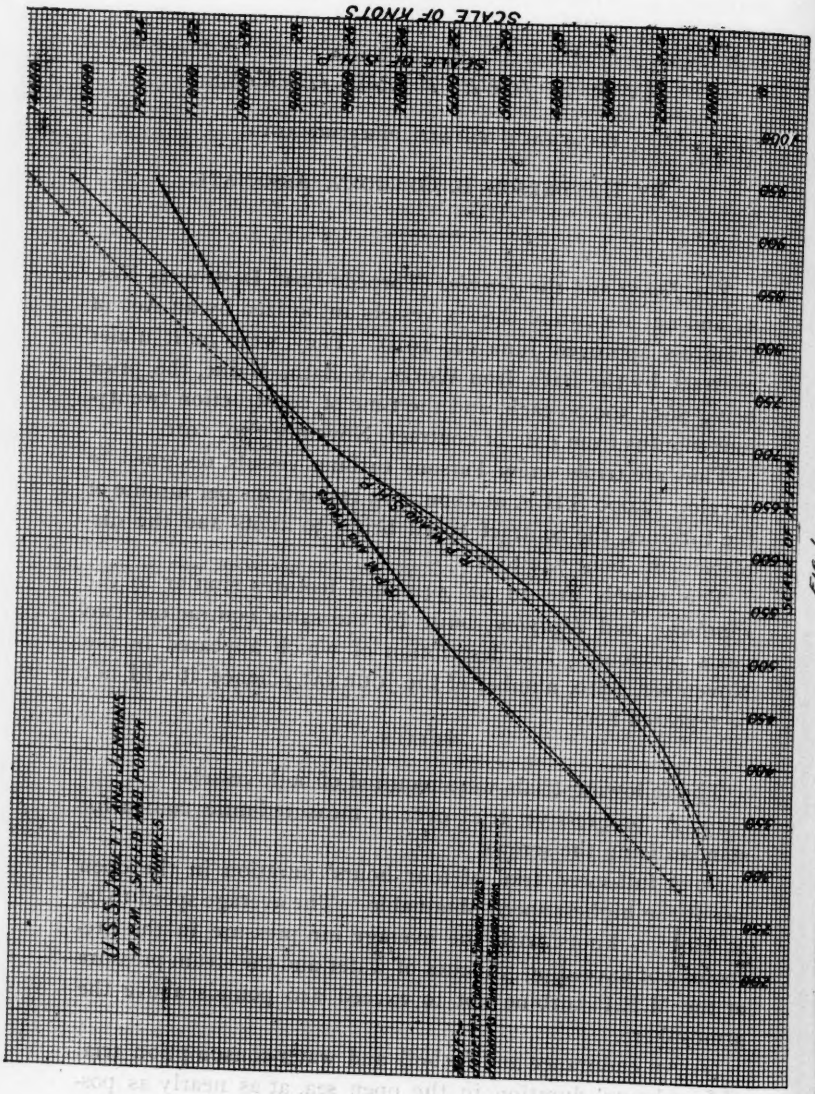

SCALE OF KNOTS
SCALE OF H.P.
U.S.S. Jouett & Jenkins
H.P. & Speed on Power Runs
Jouett
Jenkins
Full Power Trials
Full Power Trials
Scale of H.P.

FIG. 1.

FIG. 1.

TABLE I.—STANDARDIZATION TRIAL DATA U. S. S. "JOUETT."

Course off Rockland, Maine. April 30, 1912.

Number of run	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	20	21	22	23	24
Time on course, minutes. seconds.	3 50.78	3 57.3	3 58.78	3 34 3	3 22.2	3 32.4	2 45.6	2 53.6	2 46.67	2 25.7	2 20.5	2 24.4	2 8.7	2 11 8	2 9 2	2 0.1	1 58.3	1 59.7	1 49 6	1 48.8	1 49.2	1 49.6	1 48.5
Revolutions to go 1 knot:																							
Shaft No. 1	1,418.4	1,458.9	1,410.2	1,526.8	1,425.1	1,500.0	1,451.7	1,511.0	1,460.3	1,624.1	1,587.1	1,632.9	1,528.1	1,567.1	1,526.2	1,689.1	1,657.2	1,686.1	1,742.7	1,730.5	1,738.2	1,739.1	1,721.5
No. 2	1,240 6	1,273.4	1,226.6	1,325.4	1,243.7	1,310.8	1,216.4	1,273 7	1,221.6	1,404.0	1,358.3	1,402.2	1,445.8	1,483.7	1,446 7	1,521.0	1,490.9	1,513.5	1,596.0	1,583 5	1,589.4	1,589.2	1,581.4
No. 3	1,436 6	1,473.5	1,436.2	1,470 3	1,454.2	1,527.6	1,471.3	1,526.5	1,480.2	1,479.5	1,437.2	1,479.9	1,547.1	1,581.3	1,545.4	1,699.5	1,672.8	1,695.5	1,756.0	1,744 3	1,753.9	1,756.0	1,745.0
Average	1,365.2	1,401.97	1,357.7	1,441.17	1,374.33	1,446.13	1,379.8	1,437.07	1,387.37	1,502.53	1,460.87	1,505.0	1,507.0	1,544.03	1,506.1	1,636.53	1,606.97	1,631 7	1,698 23	1,686.1	1,693.83	1,694.77	1,682 63
Revolutions per minute:																							
Shaft No. 1	369.1	368.9	362.9	428.3	422.8	423.7	526.1	522.5	525.4	669 3	677.8	678.5	712.1	713.4	709.3	844.5	839.8	845.1	954.0	954.5	955.3	952.5	952.1
No. 2	322.9	322.0	315 7	371.9	369 0	370.3	440.8	440.5	439.6	578 5	580 0	582.6	673.8	675 4	672.4	760.5	755.5	758 7	873.7	873.4	873.5	870.4	874.5
No. 3	373.9	372 6	369.6	412.6	431.4	431.5	533.2	527.9	532.6	609 7	613.7	614.9	721 0	719.8	718.2	849.7	847.7	849.9	961.3	962.2	963.9	961.7	965.0
Average	355.3	354 5	349.4	404.3	407.7	408.5	500.0	497.0	499.2	619 2	623.8	625.3	702.3	702.9	700 0	818.2	814.3	817.9	929.7	930.0	930.9	928.2	930.5
Speed, in knots	15.60	15.18	15.46	16.80	17 80	16.95	21.74	20.74	21.60	24.71	25.62	24.93	27.97	27.31	27.86	29.98	30.43	30.08	32.85	33.09	32.97	32.85	33.18
S.H.P., Shaft No. 1	364	374	358	621	590	580	1,166	1,115	1,150	2,545	2,670	2,636	2,571	2,595	2,658	3,349	3,354	3,352	4,150	4,204	4,181	4,117	3,985
No. 2	199	200	189	326	312	313	481	492	491	1,436	1,529	1,414	2,720	2,708	2,714	3,112	3,164	3,145	3,516	3,515	3,539	3,496	3,398
No. 3	389	408	375	510	628	604	1,260	1,193	1,273	1,859	1,821	1,824	3,008	3,026	2,941	4,180	4,146	3,993	5,304	5,341	5,297	5,338	5,356
Total, all shafts	952	982	922	1,457	1,530	1,497	2,907	2,800	2,914	5,840	6,020	5,874	8,299	8,329	8,313	10,641	10,664	10,490	12,970	13,060	13,017	12,951	12,739
Number of turbines used.	Five.									Four.			Three.										

TABLE II.—STANDARDIZATION TRIAL DATA U. S. S. "JENKINS."

Course off Rockland, Maine. May 21, 1912.

Number of run.....	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
Time on course, min sec.	4 45.3	4 26.8	4 49.72	3 38.2	3 47.9	3 39	3 3.5	2 59.8	3 4.5	2 35.1	2 35.8	2 35.5	2 22.95	2 24.5	2 21.7	2 9.9	2 7.9	2 9 7	1 58.75	1 59.2	1 57.5	1 49.75	1 48.95	1 49.15	1 48.7	1 48.8
Revs. to go 1 knot :																										
Shaft No. 1.....	1,493.1	1,397	1,489.9	1,433.5	1,493.8	1,436.2	1,507.5	1,469.5	1,503	1,553.1	1,558.5	1,558	1,606	1,618.4	1,598.4	1,563	1,546.5	1,567.2	1,670.4	1,682.8	1,661.8	1,753.5	1,724.3	1,756.9	1,735.3	1,755.7
No. 2.....	1,293.9	1,198.9	1,283	1,229.7	1,285.7	1,235 5	1,215	1,193.1	1,223	1,322.7	1,332.5	1,328.2	1,361.7	1,374	1,356.4	1,472	1,453.7	1,470.5	1,506	1,515.1	1,498.9	1,611.9	1,585.6	1,617.3	1,598.9	1,613.4
No. 3.....	1,504.6	1,392.5	1,499.8	1,424.6	1,494.9	1,430.1	1,518.6	1,464.1	1,510.4	1,418	1,436	1,424.3	1,446.6	1,450 8	1,441.2	1,578.4	1,577	1,581.4	1,704	1,705	1,693.8	1,775.1	1,757.9	1,783.6	1,768.3	1,785.2
Average.....	1,430.5	1,329.5	1,424.2	1,362.6	1,424.8	1,367.3	1,413.7	1,375.6	1,412.1	1,431.3	1,442.3	1,436.8	1,471.4	1,481.1	1,465 3	1,537.8	1,525.7	1,539.7	1,626.8	1,634.3	1,618.2	1,713.5	1,689.3	1,719.3	1,700.8	1,718.1
Revs. per minute :																										
Shaft No. 1.....	314.01	314.29	308.56	393.98	393.4	393.56	492.92	490.46	488.79	600.18	599.81	601.01	679.06	671.99	676.33	721.78	725.09	725	843.83	846.71	848.57	958.4	948.9	965.54	957 16	968.45
No. 2.....	272.11	269 71	265.74	337.99	338.55	338.58	397.29	398.21	397.74	510.77	512.82	512.38	571.53	570.53	573.93	679.39	681.56	680.27	760.76	762.34	765.39	881.03	872 63	888.79	879 39	889 95
No. 3.....	316.43	313.28	310.61	391.56	393.66	392.89	496.55	490.31	491.17	547.66	552.68	549 47	607.2	602 4	609.83	728.92	739.39	731.57	860.63	857.86	864.91	970.24	967.45	980.22	975.61	984 73
Average.....	300.85	299.09	294.97	374.51	375.2	374.68	462.25	459 66	459 23	552.87	555.1	554.28	617.59	614 97	620.03	710.03	715.35	712.28	821.74	822.30	826.29	936.56	929.66	944.85	937.39	947 71
Speed in knots.....	12.618	13.493	12 427	16.497	15.796	16.438	19.619	20.022	19 512	23.210	23 107	23.151	25.183	24.913	25.406	27.714	28.147	27.756	30.316	30.201	30.638	32.802	33.042	32.982	33.119	33 088
S.H.P., Shaft No. 1.	276	277	272	520	530	520	1,058	1,026	982	2,048	2,013	2,034	2,782	2,773	2,791	2,880	2,952	2,872	3,900	3,937	3,876	4,746	4,699	4,782	4,793	4,876
No. 2.	141	139	130	257	267	267	340	379	379	986	1,004	1,003	1,430	1,443	1,436	2,679	2,688	2,683	3,497	3,463	3,497	4,170	4,059	4,085	3,899	3,800
No. 3.	239	237	226	472	496	483	1,043	1,043	990	1,303	1,285	1,262	1,785	1,704	1,776	3,123	3,147	3,113	4,049	4,131	4,214	5,243	5,174	5,242	5,053	5,322
Total, all shafts.....	656	653	628	1,249	1,293	1,270	2,441	2,448	2,351	4,337	4,302	4,299	5,997	5,920	6,003	8,682	8,787	8,668	11,446	11,531	11,587	14,159	13,932	14,109	13,745	13,998
No. turbines used...	Five.									Four.						Three.										

TABLE III.—TRIAL DATA.

.....	U. S. S. *Jouett*.			U. S. S. *Jenkins*.		
	4-hour full-power trial.	4-hour 25-knot trial.	4-hour 16-knot trial.	4-hour full-power trial.	4-hour 25-knot trial.	4-hour 16-knot trial.
Date of trial	1912. April 30.	1912. May 1.	1912. May 1.	1912. May 21.	1912. May 22.	1912. May 22.
Speed, in knots	32.266	25.14	16.008	31.272	25.055	16.031
Slip of propellers, per cent. of own speed, mean:						
Starboard propeller	26.58	21.17	13.55	25.74	21.33	13.54
Middle propeller	19.87	9.48	2.58	18.74	8.49	1.59
Port propeller	27.66	13.51	13.03	27.13	13.09	13.36
Draught, mean on trial, feet and inches	8–00 $\frac{31}{32}$	8–02 $\frac{3}{16}$	8–03 $\frac{7}{32}$	8–00 $\frac{17}{32}$	8–02 $\frac{8}{8}$	8–02 $\frac{25}{32}$
Displacement corresponding, tons	728.5	730.5	723.3	718.9	726	721
Number of turbines used	3	4	5	3	4	5
boilers used	4	4	1	4*	2	1
oil burners used (8 per boiler)	31	24	7	32*	16	5
Heating surface used, square feet	18,000	18,000	4,500	18,000*	9,000	4,500
Pressures (average):						
Main steam, at boilers, pounds gage	247.5	259	177.5	227.5	252.5	180
steam chest, pounds gage	235†	240‡	132.5§	212†	235‡	115§
Vacuum in condensers, inches of mercury	27.6	29	29	27.5	29	29.2
Air pressure in fireroom, inches of water	4	1.48	1.8	4.95	4.3	3
Revolutions per minute of main turbines (average):						
Shaft No. 1	921.4	668.61	388.24	882.95	667.99	388.74
No. 2	844.3	582.33	344.45	806.84	574.45	341.54
No. 3	935.2	609.41	385.91	899.76	604.66	387.96
Mean, all shafts	900.32	620.12	372.87	863.18	615.63	372.75
Collective S.H.P., all shafts, from curve	12,340	5,850	1,115	12,440	5,925	1,250

* Last 2 hours and 18 minutes of trial run with 3 boilers, 24 burners and 13,500 square feet of heating surface. † M.H.P. ‡ I.P.C. § H.P.C.

sible a uniform speed of 25 knots, the average not to fall below that figure.

(*d*) An endurance and fuel-oil and water-consumption trial at 16 knots, under similar conditions to the preceding trial.

The various trials were conducted April 30 and May 1, 1912, for the *Jouett*, and May 21 and 22, 1912, for the *Jenkins*. Both standardization trials were run on the measured-mile course, Rockland, Me., and all other trials were run in the open sea off the Maine coast. Good weather prevailed on all the trials.

On the standardization trials twenty-three and twenty-six runs, at various speeds, were made over the measured mile, for the *Jouett* and the *Jenkins*, respectively, the data from which were used in plotting the curves shown in Figure 1. The data are given in Tables I and II.

From the official curves the following revolutions per minute of the main turbines were found to be necessary to attain the speeds stipulated for the other trials:

Speed, in knots.	*Jouett.* r.p.m.	*Jenkins.* r.p.m.
16	372.5	372.0
25	615.5	613.5
30	805.0	806.0

The full-speed and endurance trials were successfully run on the dates noted in Table III. The contract speed was easily exceeded on the full-speed trials of both vessels, as shown in the table, which also gives the other data obtained on the several trials of both vessels.

GOVERNMENT TESTING AND INSPECTION OF MATERIAL FOR THE NAVAL SERVICE.

By Captain Walter F. Worthington, U. S. N., Member.

The position of the U. S. Government in regard to the matter of testing and inspection of material is somewhat different from that of other Governments and also from that of any private corporation in this country, for the following reason: In the case of others receiving material of an unsatisfactory nature, it is within their power to discontinue dealings with that manufacturer, and this possibility is in the nature of a penalty continually operating to cause the manufacturer not only to deliver material complying with the letter of the specifications, but, further, to fulfill in the best manner in regard to durability and all other qualities the purpose for which it is intended, so that a customer once obtained may be held indefinitely against all competitors. The U. S. Government, however, asks for bids in the open market on all large orders and treats all bidders on a perfect equality, notwithstanding the fact that one or more of these bidders may have delivered poor material on former occasions and others may be known to be reliable and may be depended upon to deliver material uniformly good. Another restriction on the Government is the necessity of having, in some cases, high-grade material of special manufacture for war ships, which would be unnecessarily expensive for merchant ships. For example, the boilers of such ships are of the water-tube type, and about one-third the weight of such boilers is composed of tubes. If a single tube gives out the boiler is out of service until it can be replaced, and one boiler cut out means the ship cannot maneuver at full power with the fleet for that length of time.

Under such conditions a system of inspection and testing has grown up which becomes more and more exact and complete from year to year, framed for the purpose of securing the desired materials with uniform certainty, leaving no loop hole for failure either from accident or design.

The method of proceeding varies somewhat according to circumstances, and for simplicity that adopted in the Pittsburgh District will be described.

In the first place, materials and supplies for the Navy are nearly all obtained in the northeastern part of the United States, and this area is divided into inspection districts with headquarters at the principal cities, such as Boston, Hartford, Brooklyn, Philadelphia, Harrisburg and Pittsburgh. This last covers parts of ten States and 118 cities and towns.

It is a matter of interest to note that this section of the country is one of the most densely populated areas in the Western Hemisphere. Statistics collected by a local organization show that there is within a circle having Pittsburgh as a center and a radius of 450 miles, a total population, in round numbers, of 42,000,000; nearly equal to half the population of the U. S., while a corresponding circle around New York City has a population of only 31,000,000, Chicago 30,000,000 and Boston only 25,000,000. This gives a population for the circle around Pittsburgh of 264 per square mile, which is 33 per cent. greater than the density of the population of France (196), and nearly equal to that of Germany (290.4).

While this district is principally known for its manufactures of iron and steel, there is an immense variety of other material produced. Out of one hundred and ninety-two kinds of supplies required by the Bureau of Steam Engineering alone, all but sixteen are made within these limits.

Under the circumstance it is necessary to have an inspection force of a flexible character which can be expanded or contracted on short notice and the members of which are capable of handling a great variety of materials, some of them of vital importance to war ships, such as armor and boiler tubes.

The organization of the Inspection Force for the Bureau

of Steam Engineering is similar to that of the other Bureaus, and is as follows:

The main office, at Munhall, Pa., is taken as headquarters, at which is stationed a naval officer with one assistant for engineering work and one for electrical work, a chemist and two clerks. The remainder of the assistant inspectors are located at the centers where the greatest amount of work is turned out, and these centers vary from time to time, but, generally speaking, are near Pittsburgh or else nearly on a line between Cleveland and Cincinnati.

Communication with these inspectors is by letter, telephone or telegraph, and they make daily reports by mail which show what they have done and what they are about to do; also whether they have received papers and orders addressed to them.

Orders to inspect come to headquarters from the various Bureaus of the Navy Department and also from the purchasing officers under that Department located at the various naval stations and cities where work for the Department is being done. These orders are accompanied by a copy of the contract giving the specifications and the items referred to. Upon the receipt of such an order it is stamped with an identification file number for convenience of reference, and a letter is at once despatched to the manufacturer requesting a copy of his order, which, of course, should be a duplicate of the other but often is not, owing to mistakes and omissions in copying. The same letter requests that the office be notified when the material is ready for inspection. The manufacturers' order, when received, is compared with that obtained from Washington, and mistakes, if any, corrected before much, if any, trouble has arisen. This comparison is a great help to the manufacturer.

When notification is received from the manufacturer that the material is ready for inspection the fact is entered at once in a book kept for the purpose, giving the date of receipt of the notification and the date when the material was ready. The assistant inspector nearest the place of manufacture is notified and furnished with all the papers he requires, giving him the items, specifications and any special instructions.

Upon the completion of an inspection, if satisfactory, the assistant places his personal stamp and a Government stamp on the material and authorizes the company to make shipment, and requests them to furnish to headquarters information as to the number of the railroad car, the name of road and date when car was dispatched from their works. He also sends to headquarters his report of the results of the inspection. When this latter is received a letter is addressed to the manufacturers asking for the information which the Government inspector requested, this as a reminder and a record in case the manufacturer may forget to send the information or might claim that he did not know it was wanted. When the information is obtained as to the car number, etc., that and a complete description of the material referred to is mailed to the navy yard or station where delivery is to be made, and also to the Bureau of the Department concerned. Upon the receipt of the material and of the paper describing it the Board of Inspection at the naval station is in a position to know just what inspection has been made and to check up every item. This Board makes any further inspection thought desirable, and in case it finds defects not previously discovered by the first inspector they are at liberty, under the terms of the contract, to reject all or any part of the consignment or to require the manufacturer to make good deficiencies.

The foregoing gives in a general way the method of procedure, but there are many details which vary according to the character of the material. For example, in the case of Boiler Plate.

BOILER PLATE.

The first thing for the inspector to do preliminary to making an inspection is to familiarize himself with the equipment and mill practice of the manufacturer, as in many cases he thus gets information that enables him to direct special attention to certain parts of the material, which under the circumstances, are likely to be defective. It is not always practicable for the inspector to witness the whole process of manufacture, and when not practicable he relies on the records kept by the manu-

facturer, and which are accessible to him. From them he gets the heat number, location of the slab in the ingot, number of slabs rolled, etc. The inspector, however, is always notified in time so that he can be present when the slab is rolled. If he cannot arrive in time to see the rolling, he must be there immediately after the plate has been laid off. He first inspects the top surface of the plate for such defects as are to be found, such as cinder pits, excessive scale, marks, seams, snakes; blisters and burnt edges. If the plate passes this preliminary inspection he indicates where the test pieces are to be taken and stamps them and the plate after the slab number has been stamped in each case. Two longitudinal tensile test pieces and two transverse bending-test pieces (one of each for spare) are selected, the latter from one end of the plate and the former from the side and near the opposite end. No further inspection is made until the test pieces have been machined to shape. Upon receiving notification that the test pieces are ready, he first identifies them by their stamps and balances the testing machine.

While the pieces are being pulled by the operator the inspector watches the operation closely to get his own independent readings, regardless of those taken by the operator.

The bending test is made under a steam hammer or hydraulic press, if available. Drillings are taken by the inspector from the bending-test pieces and analyzed by the Government chemist. If the analysis proves satisfactory the final inspection is made.

After the plate is identified by the numbers and stamps previously put on it it is hoisted high enough to allow the inspector to walk under it and make an inspection with an electric light or torch. The defects looked for are the same as those on the top surface, and in addition he looks for brick and burnt surface. The plate is then lowered to a convenient height and the edges examined for splits, and it is then gaged at the four corners where a deficiency in this respect is most likely to occur. The weight is then taken, and if all is satisfactory it is stamped for shipment. The shipping report describing the

plates, giving the number of the car, date of shipment, etc., is forwarded to the officer to whom the material is consigned, in a manner similar to that to be described in the case of boiler tubes.

BOILER TUBES—COLD-DRAWN, SEAMLESS.

The specifications require the following tests:

(1) A piece from each end of each tube to stand flattening without showing any defects on the outside.

(2) Ten per cent. of these flattened pieces when opened out to show no defect on the inside.

(3) Two per cent. of tubes annealed in the same furnace at the same time must stand expanding to 1 1-16 times the original diameter and have a flange turned all around the end of the tube, equal in width to one-eighth of the diameter of the tube, without showing defects.

(4) Each tube must stand a hydrostatic pressure of 1,500 pounds per square inch.

(5) Each tube must then pass a close surface inspection inside and out and gaging, and if 10 per cent. of a lot fail on this test the whole lot may be rejected. A tolerance of 6 per cent. below the specified thickness is allowed on gaging.

(6) The tubes are to be weighed in lots, and a tolerance of 5 per cent. overweight and 3 per cent. underweight will be allowed (the weight to be calculated on a basis of 0.283 pound per cubic inch for steel). A pile of tubes, estimated enough for a day's work, is set aside, and from it tubes are taken one by one. First a piece about 4 inches long is cut from each end of each tube and at once flattened. Then 10 per cent. of these pieces, each from a different tube, are opened up. The defects looked for are broken tubes, laps, slivers and seams. While these tests are in progress a piece is cut from every 25 or 30 tubes and subjected to the expanding and flattening tests. This is to discover split tubes and the toughness of the metal. All tubes which pass these tests are then stamped for future identification and inspected for outside surface defects as follows:

They are first moved to a location where there is a good light, placing about a dozen on the trestles at one time. Measure the length and outside diameter; a tolerance of $\frac{1}{8}$-inch excess in length is allowed. Gage each end in at least four places. Examine outside surface, one-fourth at a time, turning the tube as required. Look for laps, seams, tears, guide marks, checks, rings, scratches, sinks, slivers, excessive scale, cracks, pits, burnt steel and injuries due to rough handling, and note if the tube is perfectly straight. The inside is then examined thus:

Roll the tubes, three at a time, and look inside each end, while a light is held at the opposite end, for sinks, tears, slivers, burnt metal, excessive scale, checks, laps, pits, rings and injurious scratches and piped steel. The bent tubes will not roll and are thrown out. Tubes having concentric ends but a thin wall somewhere near the middle are also detected by this rolling.

Those tubes passing this inspection are marked with a second stamp along side of that previously mentioned. Then weigh the tubes in lots, the number in a lot as found most convenient under the circumstances. Test each under the hydrostatic pressure of 1,500 pounds per square inch. If the ends of any tubes are swaged down, swelled, upset or reinforced, they are afterwards subjected to a further inspection to see that no damage has been done by these operations.

The defects to be looked for in this case are laps, fins and scale pits. After the inspection is completed the tubes are stenciled for identification and coated with linseed oil to preserve them when stored. They are then loaded into a car, counted in place, securely blocked and tin tags affixed to the inside of both car doors, the doors sealed, and seals stamped and a notice placed on the outside of the door directing that notice be sent to the proper authority in case it is necessary to break the seals before arriving at destination (as in case of damage to the car). When the car leaves the works, the date of leaving, name of railroad, and all information necessary to identify it, is at once reported to the inspector of the district,

who forwards a copy for the inspector at the point of destination, who will not receive the tubes until he gets this notice that this inspection is satisfactory in all particulars.

Finally, if, after this close inspection any of the tubes are found defective when being put into boilers at the builders' yards, such tubes are rejected and replaced by the manufacturer. A report from the inspector at the building yard, of the number of such rejections is received by the inspector of the tubes, each month, with a notation as to the cause of rejection. The inspector then traces up the name of the assistant who stamped the tubes and attempts to discover the exact cause of the oversight and to provide a remedy to prevent its recurrence. It may be added that the percentage of tubes rejected at the builders' yards is now very small indeed.

This whole system has been built up with the hearty cooperation of the tube manufacturers, who find it a far better plan to discover defects at the mills rather than at any subsequent stage.

It may strike some of those present that the Government is taking extreme and perhaps unnecessary precautions to insure that none but absolutely perfect tubes shall be put into the boilers, and also, that if many of the apparently unimportant defects enumerated were waived it would be possible to greatly reduce the cost of the tubes.

Those who have had any experience in the stokehold of a war vessel need no explanation or apology for the apparently hypercritical inspection made of the tubes.

For the benefit of the others it may be mentioned that there are two reasons for this extreme care. One is for the sake of the men and the other for the ship.

In a large number of naval vessels, especially of the smaller ones, such as destroyers and torpedo boats, the stokehold is made of the least width in which it is practicable to work a slice bar, the object, of course, being to crowd into the vessel the greatest possible amount of boiler power. This stokehold is made airtight, and to accomplish this it is necessary to have all doors securely fastened. This latter means that it takes

an appreciable time to open the doors and escape, in case of an explosion. If a single tube gave out a large quantity of steam will escape; to prevent this from entering the stokehold, automatic ash-pit doors and furnace doors are provided, but engineers all know that automatic apparatus of this kind sometimes fails to work; also it is necessary to open these doors very frequently to coal the furnace, remove the ashes, etc., much more frequently when steaming full speed, and therefore at the time when the tubes are most likely to go. Although the men who enlist in the Navy never show the slightest hesitation in going wherever duty calls, it is only fair to them that those who plan the ships should take every practicable precaution to prevent their lives being sacrificed unnecessarily.

With regard to the ship the failure of a single tube in a water-tube boiler, such as are used almost exclusively in the Navy, means a great deal more than the failure of a tube in the Scotch boiler used for mercantile ships. In the latter case the tubes can usually be plugged temporarily while under steam, and the boiler soon got ready for full power again. In the former case the boiler must be completely emptied of steam and water, the tube cut out and either replaced by a new one or else the opening in front and back header closed with screw plugs.

TUBE RUPTURES.

Tube ruptures in water-tube boilers cause serious personal injuries or loss of life, entail considerable property damage and occur all too frequently.

Such ruptures are due either to material that was originally defective or which subsequently became defective because of being subjected to continual heat, or which became defective by pitting or corrosion, or which became weakened by overheating, due to low water or to the forcing of the boiler too far beyond its rated capacity.

When the rupture occurs the fireman has to suffer, and frequently the rupture of a single tube allows all of the water

to escape from the boiler, as a result of which the entire boiler is damaged beyond repair.

Tubes in water-tube boilers should be carefully inspected and thoroughly hammer-tested to determine their condition; and whenever there is reason to believe that the tube material has lost its ductility by reason of its long service in the heat of the furnace, or is otherwise defective, the tube should be replaced by a new one having proper strength and ductility.—(Monthly Bulletin of the Fidelity and Casualty Company, quoted by "The Boilermaker," June, 1912.)

TROUBLES OF INSPECTORS.

When the present system of Government inspection was first established, about the time when it was decided to build a "new Navy" of modern ships, some of the manufacturers had grave misgivings as to the effect the system would have on them. They expected that the inspectors would find out their secrets of manufacture and disseminate the information among their rivals. This danger was partly obviated by detailing no one but naval officers for this work, it being thought they would be less tempted to betray secrets and also under better control than civilians taken on temporarily and who, having no permanent position or prospects of promotion, would be likely to accept any good offer of employment from a manufacturer. Even this precaution, however, did not satisfy one builder of a torpedo boat, and he was so insistent that the Navy Department allowed him to proceed without an inspector in the works and with the sole condition that the finished boilers and machinery were to be inspected by an officer. In due course the builder reported that the boilers were ready for inspection. The officer detailed for the purpose entered the works and asked what material the boiler tubes were made of, the builder proudly stated that the tubes were made of the best Swedish iron. Thereupon the officer informed him the boilers were rejected. The Act of Congress providing money for the purchase of the boilers, distinctly specified that all

materials were to be of American manufacture. The contractor should have known this, but overlooked it.

Since that time the leading manufacturers have changed their views a good deal, as they find, from time to time, instances where the inspectors forestall mistakes and oversights.

This is particularly the case with boiler tubes, the inspection of which, though so very rigid, goes on most smoothly, the specifications having been revised again and again, made more definite each time, although more rigid, but all with the full coöperation of the manufacturers and in the light of an intimate knowledge on the part of the Government officers, of what can be done in every-day practice and with a reasonable amount of certainty.

This is a most striking example of the wisdom of the manufacturers in allowing the Government officers to get thoroughly acquainted with every detail of the process. The latter can see for themselves when a requirement is unreasonable and understand when the manufacturers wish some change allowed. If all the other manufacturing processes were understood as well, there would be a great improvement in other specifications.

The best way to reduce the number of troubles with which the inspector has to contend is to speak out and say exactly what they are, noting that some of them can be remedied by the Society for Testing Materials. Troubles may be caused by (1) the contractor; (2) the salesman employed to represent the manufacturer; (3) by the specifications which form part of the contract; (4) and most of all, by the middle man, who knows very little about the goods or the uses to be made of them.

It ought to be possible for the Government to advertise for bids on any article whatever and under any specifications whatever, provided that the specifications clearly and accurately describe what is wanted.

Many bidders, however, do not see this, but make their bids at random, on the grounds that the article they produce is good and is bought by many persons, and therefore ought to

be used by the Government, and assume that the Government does not know what it wants.

When the inspector rejects such an article a long argument is begun as to whether or not the said item would answer the purpose or whether or not it would not even be better than the one described, since it is made to conform to the standard specification used by some other branch of the Government or by some large corporation.

They lose sight of the fact that the other bidders made a price on a definite proposal, and if one is allowed to supply a substitute then all other bidders should be allowed to revise their first price; in other words, the items should be re-advertised.

But this procedure would cause a delay in the receipt of the article by the Government and would be unjust to the Government.

The form of Government contract describes in detail what is wanted, when it is required, etc., and provides that a bidder may submit an alternative bid, the understanding being that if the alternative bid covers in a satisfactory manner what is wanted, and the terms are most favorable it will be accepted. This is a fair proposition, if carried out faithfully by both parties. It happens, however, that a bidder will submit an alternative proposition and, after the article is reported by him to be ready for inspection, it is found not to conform to either the original specifications or his own specifications. For example, when a certain high-grade lathe was called for, the manufacturer submitted an alternative bid describing a lathe he proposed to furnish, which, he said, was 12 inches shorter, but omitted to say had less swing.

When presented for inspection, it was found, to be shorter, as he stated but had a swing $2\frac{1}{2}$ inches less than described in his alternative bid, and when it was rejected, he protested on the grounds that the deficiency in the swing was a "mere technicality," and added that the smaller lathe would last just as long as the larger one. What he did was simply to submit a smaller lathe of inferior finish in respect to details.

Another trouble arises in the very beginning from the over-eager sales agent of the manufacturer signing a contract for his firm to supply an article which they have not the appliances for manufacturing. He either doesn't read the specifications, or does not understand them, or else doesn't know exactly what his firm can do.

For example, a firm contracted to deliver certain steel floor plates with ribs of square section. When the plates were reported ready, it was found that the ribs had a curved section, and the firm had no rolls with which to make them otherwise. To provide such rolls would have taken up all the profits; also they had no straightening rolls and wanted to deliver the plates warped and twisted, just as they came from the " hot beds." When questioned about the matter, the superintendent of the mill said he had never seen any copy of the specifications.

In another case, a firm undertook to furnish in a hurry, two 9-inch engine shafts of high-grade steel, annealed. They had no hammer large enough for a shaft of that size and no annealing furnace nor any men with experience in such work. After several trials they completely failed, as was anticipated by those familiar with that kind of work. The same firm had failed the year before on similar but smaller work.

The obvious remedy for such troubles is for manufacturers to employ as sales agents only men who are engineers and also of some practical experience; men who will read specifications, who are able to understand specifications and who will take the trouble to keep themselves fully posted with regard to their firm's shop equipment and capabilities.

As an illustration of difficulties arising from a misunderstanding of the specifications the following may be taken:

The specifications for the stems of low-pressure globe valves were " Stem, rolled manganese bronze, phosphor bronze, or naval brass if called for by the order."

Two manufacturers in widely separated parts of the country claimed that they understood that sentence to mean they had an option in every case of using any one of the three metals.

It, of course, only admits of that construction if the words

" if called for by the order" are ignored and held to have no meaning at all, or by inserting a comma before the word " if," etc.

This method of misreading a sentence could, of course, be applied to any document, causing the meaning to be changed completely; but it is not a legitimate or reasonable method of construing. However, to avoid further discussion, the sentence was changed to read " Stem, rolled naval brass if so stated on the order, otherwise of rolled manganese or phosphor bronze."

The moral of all this is, it is better in writing specifications to divide a sentence up into a number of short ones, placing no reliance in such little matters as punctuation marks. It is preferable to have the printed copy look like poetry, rather than to have it misunderstood like poetry.

CHEMICAL COMPOSITION OF BRASS.

One of the greatest sources of trouble and delay in obtaining articles of composition material is the failure of manufacturers to meet the chemical requirements.

As it would be inconvenient, and in the case of small valves, impracticable, to cast test pieces attached, and if cast separately from the same melt could not be identified, unless there was an inspector always on hand when such castings were made, it is the practice of the Government to simply require drillings to be taken by the inspector from the finished casting, and analyzed. From long experience at the navy yards in making brass castings of certain mixtures, it is known what are the physical characteristics of these mixtures, assuming that they are made properly by skilled workmen. Also, chemical analysis is simple and reliable. However, when manufacturers find their material rejected on account of excess of copper or tin, they often claim that injustice is done them. They assert that the right amount of copper and tin were put in and therefore the chemist must be mistaken, and propose to take drillings from some other part of the metal. This, of course,

is never allowed, because it is quite possible to find some part of bad material good. The contract requires that all parts must be good and the material of uniform quality. Another claim made is that the material is better than that specified, because of an excess of more expensive materials, copper and tin. This claim is also rejected, because the question of price is not the only factor. The Government is not paying for the components separately, but for the completed product. The strength of the specified composition is known, but not the strength of every other conceivable mixture. Further, the manufacturer in such cases does not submit any data taken by disinterested experts, to show that the particular mixture in the case in question is as good as that specified for the purpose for which it is to be used. Also the fact that the manufacturer attempted to produce a certain composition and ended by producing a different one shows lack of skill on his part, and throws doubt on the quality of his product, especially when the composition he produces costs him more than that he contracted to furnish. The truth is that when a composition turns out to have a larger percentage of copper or tin, or both, than called for by the contract, it is not the least likely that the manufacturer put in more of these metals than required, but it is likely that the more volatile zinc has been driven off by melting at too great heat. It may also be true that from some inequality of the mixing there is too much copper or tin in one part and too little in another, and the drillings happen to have been taken from the former part. This would mean lack of uniform strength in the different parts, and is a just cause for rejection.

The Government specifications for brass allow a liberal tolerance, and are usually met at first trial by one manufacturer in this district, but give trouble to several others.

For instance, in the case of phosphor bronze, copper may vary from 85 to 95; manganese bronze 59 to 63, etc.

When an analysis is made and it is found that the material fails to meet the specifications a check analysis is made at once. In the case of rejection on account of the chemical

U. S. S. *WYOMING.*

DESCRIPTION AND OFFICIAL TRIAL.

By H. L. Brinser, Lieutenant Commander, U. S. N., Member.

The U. S. S. *Wyoming* (Battleship No. 32) is one of two first-class battleships of 26,000 tons displacement authorized by act of Congress approved March 3, 1909, and was built by The William Cramp & Sons' Ship and Engine Building Company, of Philadelphia, Pa.

The contract, which was signed on October 14, 1909, provided for the delivery of the *Wyoming* in thirty-two months, at a cost of $4,450,000, exclusive of armor, ordnance, boats and lesser outfit usually furnished by the Government.

The contract required a speed of 20½ knots for four hours' duration in open sea, deep water, at mean trial displacement of about 26,000 tons. The keel was laid down on February 9, 1910, and the vessel launched May 25, 1911.

HULL DIMENSIONS.

Mean draught, trial, designed, feet and inches	28–6
Displacement on above draught, tons	26,000
per inch at mean trial draught, tons	88.411
Length between perpendiculars, feet and inches	554–0
overall, feet and inches	562–0
Beam, extreme, feet and inches	93–2⅝
Area of immersed midship section at mean trial draught, square feet	2,620
Coefficient of fineness, block, at mean trial draught	.618

BATTERY.

The main battery consists of twelve 12-inch 50-caliber breech-loading rifles arranged in pairs in six turrets on the

center line of the ship. There are two 5-inch by 21-inch submerged torpedo tubes.

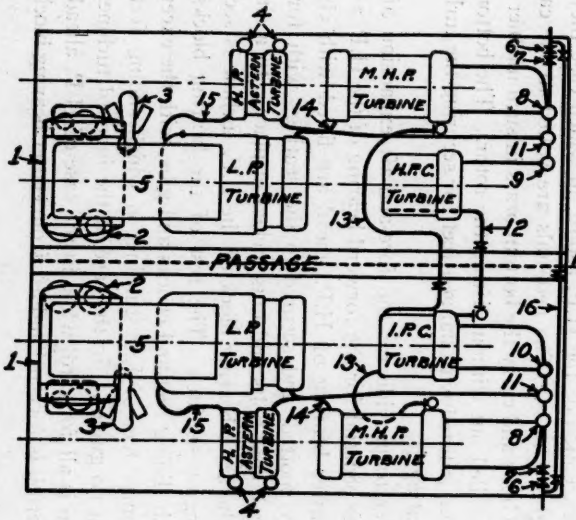
The secondary battery consists of twenty-one 5-inch 51-caliber R. F. guns, seventeen of which are in splinter-proof compartments on the gun deck, the remaining four being located on the bridge and the superstructure decks. The smaller guns comprise four 3-pounder guns for saluting purposes; two 1-pounder guns for boats; two 3-inch field pieces; two .30-caliber machine guns.

MAIN ENGINES.

The propelling machinery consists of Parsons turbines of 28,000 shaft horsepower when running at 330 revolutions per minute. They are arranged on four shafts, and are disposed as shown in Fig. I.

Bearings.—There is one main bearing at each end of each turbine. It consists of a pedestal cast separate from the casing with the exception of the H.P. astern, in which case they are cast integral with it. The pedestals are bolted to the casing with steel bolts and carry the bottom brass in such manner that it can be rolled out without lifting the rotor. The bottom of the pedestal is fitted with flanged feet for securing the turbines to the seatings.

In order to provide for the fore-and-aft expansion of the casing, the pedestal for the forward bearing of the H.P. astern and the after bearing of H.P. ahead are fitted with chocks accurately spotted, and secured to the seatings with forged-steel body-bound bolts and nuts locked in place. In a similar manner the after bearing block of the L.P. turbine is secured to the engine seatings. The rest of the bearing blocks are fitted with sliding feet and chocks and gibs, with the exception of the center feet of L.P. turbine, in which sliding feet are fitted, but no guides. In the case of the inboard turbines the expansion is all forward, while in the case of H.P. ahead and H.P. astern it is forward and aft from center bearing blocks.



1 MAIN CONDENSERS
2 MAIN AIR PUMPS
3 MAIN CIRCULATING PUMPS
4 MAIN FEED PUMPS
5 MAIN EXHAUST TRUNKS
6 MAIN STEAM STOP VALVES
7 GOVERNOR VALVES
8 MAIN H.P. THROTTLE VALVES
9 H.P. CRUISING THROTTLE VALVE
10 I.P. CRUISING THROTTLE VALVE
11 H.P. ASTERN THROTTLE VALVES
12 EXHAUST FROM H.P.C. TO I.P.C.
13 EXHAUST FROM I.P.C. TO M.H.P.
14 EXHAUST FROM M.H.P. TO L.P. AHEAD
15 EXHAUST FROM H.P. ASTERN TO L.P. ASTERN
16 MAIN STEAM CROSS CONNECTION

Fig. 1.—Arrangement of Main Turbines and Piping ("Wyoming.")

MAIN TURBINE DATA.

	Diameter.	Length.
Rotor drums:		
Main H.P., inches	74	109¾
H.P.C., inches	73	63 1/12
I.P.C., inches	72	72½
L.P. ahead, inches	101	87⅝
H.P. astern, inches	74	36¼
L.P. astern, inches	74	44½
Number of expansions:		
Main H.P. and L.P. ahead, each	6	
H.P.C., and I.P.C., each	3	
H.P. astern and L.P. astern	4	
Turbine casings, diameter, each expansion:		
Main H.P., inches	76¼, 77¼, 78½, 80½, 83, 83	
L.P. ahead, inches	111½, 115½, 121½, 128, 128, 128	
H.P.C., inches	74½, 74⅞, 75⅜	
I.P.C., inches	74¾, 75⅜, 76¼	
H.P. astern, inches	75¾, 76¾, 78, 80	
L.P. astern, inches	83, 86½, 86½, 86½	
Length of casing for each expansion and diameter noted above:		
M.H.P., inches	17¼, 17, 17 3/16, 18⅝, 19½, 20 5/16	
L.P. ahead, inches	12 13/16, 13, 15 3/16, 15¼, 15⅜, 16½	
H.P.C., inches	18½, 18½, 25¾	
I.P.C., inches	23½, 23½, 24⅜	
H.P. astern, inches	9¼, 8¾, 8¾, 9	
L.P. astern, inches	9¾, 11 5/16, 10 15/16, 13⅛	
Rows of blading for each expansion:		
M.H.P.	13	
L.P. ahead	3 of 7 and 3 of 6	
H.P.C.	20	
I.P.C.	18	
H.P. astern	6	
L.P. astern	5	
Length of blades for each expansion:		
M.H.P., inches	1⅛, 1⅝, 2¼, 3¼, 4½, 4½	
L.P. ahead, inches	5¼, 7¼, 10¼, 13½, 13½, 13½	
H.P.C., inches	¾, 15/16, 1 3/16	
I.P. C., inches	1⅜, 1 11/16, 2⅛	
H.P. astern, inches	⅞, 1⅜, 2, 3	
L.P. astern, inches	4½, 6¼, 6¼, 6¼	
Tip clearance of blades for each expansion (designed):		
M.H.P., inches	.040, .045, .05, .055, .06, .06	
L.P. ahead, inches	.06, .07, .08, .10, .10, .10	
H.P.C., inches	.035, .04, .045	
I.P.C., inches	.04, .045, .05	
H.P. astern, inches	.04, .045, .05, .055	
L.P. astern, inches	.06, .07, .07, .07	

	Length, inches.	Diameter, inches.	Diameter, axial, inches.	Length overall of rotor drum and shaft, feet and inches.
Rotor shaft and bearings:				
M.H.P.	18¼	14	9 and 8	21–04
L.P. ahead	30	15½	8, 9, 10, 11	26–00½
H.P.C.	14½	14	9 and 11	15–03¼
I.P.C.	14½	14	9 and 11	16–0½
H.P. astern	12½	14	8	11–03¼
L.P. astern	30	15	With ahead.	

Thrust bearings:	M.H.P.	L.P.	H.P C.	I.P.C.
No. of collars	17	17	8	8
Thickness, inches	0¾	0¾	0¾	0¾
Space between, inches	$1\frac{1}{16}$	$1\frac{1}{16}$	$1\frac{1}{16}$	$1\frac{1}{16}$
Outside diameter, inches	18¼	18¼	18¼	18¼
Inside diameter, inches	12½	12½	12½	12½
Bearing surface, square inches	2,190.5	2,190.5	1,030.8	1,030⅛
No. of shoes, ahead	17	17	8	8
astern	16	16	7	7
Thrust shoes, effective thrust surface, ahead, square inches	941.8	941.8	443.2	443.2
Thrust shoes, effective thrust surface, astern, square inches	886.4	886.4	387.8	387.8

Number and diameter of steam and exhaust connections, each turbine:

	H.P.	H.P.C.	I.P.C.	H.P. astern.	L.P.A.	L.P. astern.
Steam, inches	2 of 10½	1 of 10½	2 of 10½	1 of 13	44	29
Exhaust, inches	1 of 44	1 of 18	2 of 18	1 of 29	53.47 sq. ft.	

THRUST BLOCKS.

The forward section of each shaft, with the exception of the H.P. astern, is fitted with a thrust block to take the propeller thrust and also for adjustment of clearances. The thrust bearings are carried on the same pedestals as the main bearings, and consist of semicircular bushings of Class B wrought-steel forgings grooved out and fitted with rings, in halves, of Parsons anti-friction metal (copper 6 parts, tin 1 part, and lead ¼ part). The lower half of each bearing is for taking the ahead thrust and the upper half the astern thrust. The lower bushing is so fitted that it can be rotated and removed without lifting the shaft. The bushing is fitted with a rib on its back which fits into a corresponding recess in the pedestal with ¾-inch fore-and-aft clearance on either side between the edge of the recess and the edge of the rib. Adjust-

ing rings of composition H are fitted into this slot for adjusting the amount of dummy clearance and to prevent any forward movement of the rotor. The upper part of the thrust block is fitted with a rib in a similar manner as the lower block, but is made with a tight fit to the slot in the bushing, and rigidly secured to the top cap by means of keep screws. The bolt holes in the flanges of the top cap are elongated in a fore-and-aft direction, and the whole cap is capable of fore-and-aft adjustment.

Micrometer gages are fitted on the ahead turbines for measuring the clearance.

STEAM GLANDS.

Steam glands are fitted at each end of the turbine casings. The gland casings are in halves, with flanges for securing to the turbine and to each other. The gland casings carry the gland sleeves, which are fitted so that the lower half can be rolled out without lifting the rotor shaft. The exterior of the shaft and the interior of the gland sleeves carry the labyrinth packing, in addition to which four split brass rings are fitted into grooves cut into the rotor shaft and press against the gland sleeve. The cruising turbine glands are fitted with a double steam pocket. Steam for the glands is taken from a branch of the auxiliary steam line in the forward part of the engine room, and the pressure is regulated by a reducing valve. The leak-off connections are so fitted that in case the leak off from the H.P. turbines is excessive the excess pressure can be diverted into the L.P. steam belt through a spring-loaded regulating valve. It is also arranged that any excess steam can be diverted into the turbine-drain well in the base of the L.P. casing and thence to the air pump.

SHAFTING.

There are four lines of shafting, the inboard shafts being parallel to the fore-and-aft line of the ship, the outboard shafts converging toward the center, the rake per foot being .09375 inch. All four shafts slope downward aft.

Each inboard shaft is in four sections, consisting of two line shafts supported by three spring bearings, one stern-tube shaft supported by two stern-tube bearings, and one propeller shaft supported by the strut bearing. The outboard shaft is made in two sections, one, the line shaft, supported by one spring bearing, and the propeller shaft extending through the stern tube and supported by two stern-tube bearings and the strut bearing.

The inboard coupling consists of a sleeve fitted to the stern-tube shaft by means of four $1\frac{1}{2}$-inch $\times$ 1-inch Class A steel keys. Aft of this sleeve is fitted a Class A steel collar made in halves and secured to the sleeve and disc on line shaft by means of $2\frac{3}{8}$-inch fitted steel bolts.

The outboard coupling consists of a Class A steel coupling collar fitted over the coupling discs of the shaft and secured by a sleeve, cut in halves, keyed to the shaft by two keys and bolted together by sixteen Class A steel bolts. All the stern-tube and strut bearings are lined with lignum-vitae and the shafts at these bearings are fitted with composition bushings.

DATA FOR SHAFTING.

Line shaft, diameter outside, inches	12⅛
axial hole, inches	8
journal, inches	12¾
Stern-tube shaft, diameter outside, inches	12⅜
axial hole, inches	7¼
Propeller shaft, diameter outside, inches	12⅜
axial hole, inches	7¼
Strut bearing, diameter, inches	14⅝
length, inches	72
Stern-tube bearing, diameter, inches	14⅝
length, after bearing, inches	60
forward bearing, inches	48

TORSION METERS.

Each shaft is fitted with a Gary–Cummings Torsion Meter. A description of this instrument is contained in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, No. 2, Vol. XXIV.

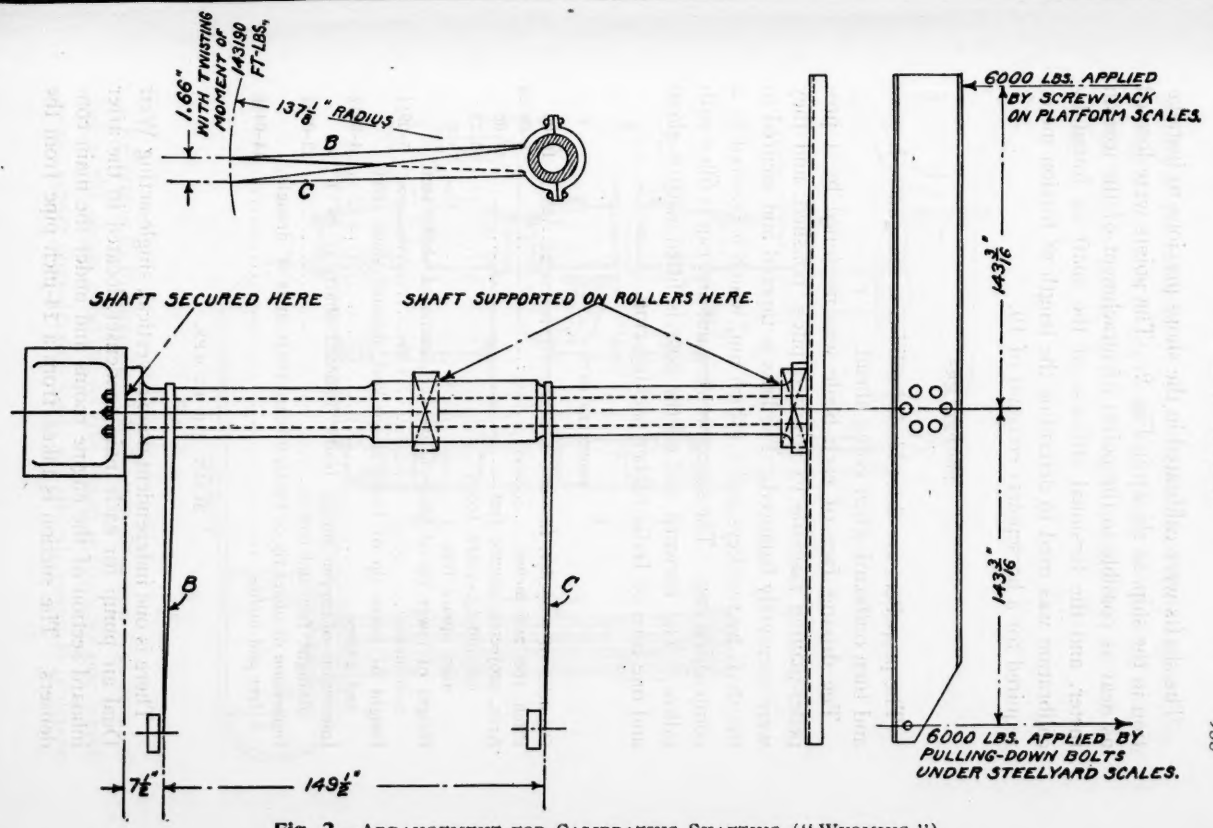


Fig. 2.—Arrangement for Calibrating Shafting ("Wyoming.")

The shafts were calibrated in the shops previous to installation in the ship, as shown in Fig. 2. The points were located as near as possible to the points of attachment of the torsion meter, and the torsional stiffness of the shaft as found by calibration was used to determine the length of torsion meter required for a horsepower constant of 10.

PROPELLERS.

The propellers are three-bladed solid-cast manganese-bronze and turn outboard when going ahead.

The driving face of each blade was machined by a propeller-planing machine to required pitch, polished, and they were accurately balanced. The boss is tapered and secured to the shaft by two keys and a 9-inch nut, which is covered by a composition cap. The space between nut and cap is filled with tallow. The forward end of the boss is fitted with a gland and one turn of India rubber packing ring.

PROPELLER DATA.

Diameter of propeller, feet and inches	10–00
Pitch, feet and inches	8–02.24
Area, projected, square feet	41.06
helicoidal, square feet	46.13
disc, square feet	78.5
Height of lower tip of blade above keel, outboard shaft, feet and inches	5–09¼
Height of lower tip of blade above keel, inboard shaft, feet and inches	4–09⅝
Immersion of upper tip of blade, outboard shaft, at 28′ 6″ draught, feet and inches	13–08¾
Immersion of upper tip of blade, inboard shaft, at 28′ 6″ draught, feet and inches	14–04⅜

MAIN AIR PUMPS.

There is one independent double, vertical, single-acting Weir Dual air pump for each main condenser, located in the after inboard section of the engine rooms and under the main condensers. The suction is taken from a 14-inch pipe from the

bottom of the condenser. This pipe is fitted with a gate valve, beyond which it is divided into two pipes, one 12-inch suction to the wet-air pump, and a 7½-inch suction for the dry-air pump.

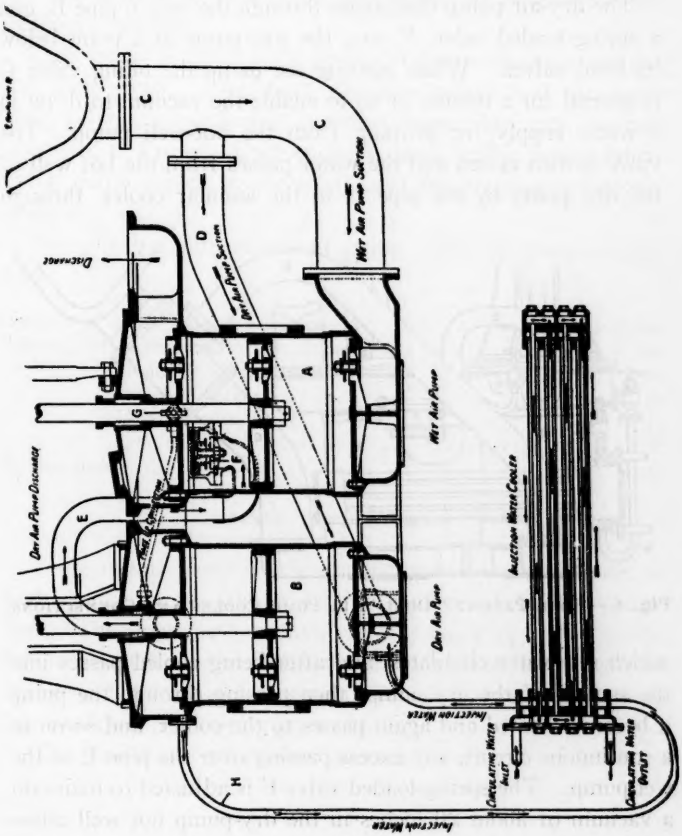


Fig. 3.

Referring to Figures 3 and 4, the wet-air pump A is situated below the steam cylinder, as this pump is the only one which works under considerable load; the dry-air pump B is driven

by a beam in the usual manner. One connection, C, is made to the condenser, but a branch pipe, D, is led to the dry-air pump, the connection being made in such a manner that the water from the condenser will pass it to the wet pump.

The dry-air pump discharges through the return pipe E, and a spring-loaded valve, F, into the wet pump at a point below its head valves. When starting the pump the filling valve G is opened for a minute or so to enable the vacuum to draw in a water supply for priming from the hot-well pump. The valve is then closed and the water passes from the hot well of the dry pump by the pipe H to the annular cooler, through

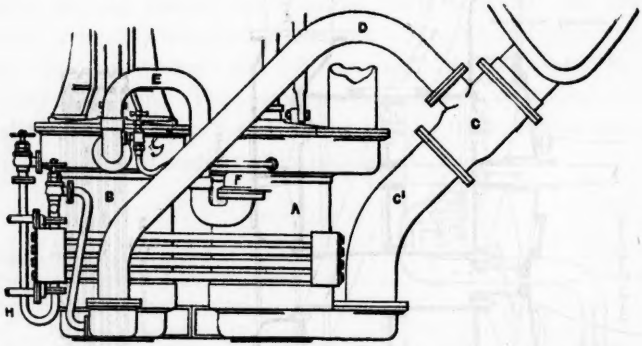


Fig. 4.—WEIR PATENT "DUAL" AIR PUMP, COOLER AND CONNECTIONS.

which sea water circulates, and after being cooled passes into the suction of the dry pump; then passing through the pump it becomes heated and again passes to the cooler, and so on in a continuous circuit, any excess passing over the pipe E to the wet pump. The spring-loaded valve F is adjusted to maintain a vacuum of about 20 inches in the dry-pump hot well when the condenser is working at 28 inches vacuum, and this difference of 8 inches in vacuum is sufficient to overcome the friction in the cooler and to pass the water into the suction. The function of this cooling water is to reduce the temperature and, accordingly, the volume of the air handled by the dry-air pump.

MAIN CIRCULATING PUMPS.

There is one double-inlet centrifugal circulating pump for each condenser, capable of discharging 15,000 gallons of water per minute under a head of 25 feet, driven by an independent compound engine.

Each pump is fitted with suction connections from the sea and main drain. The main injection valves and the main drain valve are interlocked in such manner that only one can be opened at a time. Each engine is fitted with a forced-lubrication pump driven by the shaft, and the housing is entirely inclosed.

The principal dimensions of pump and engine are as follows:

Capacity of pump, gallons per minute	15,000
Diameter of suction nozzle (2), inches	21
discharge nozzle, inches	29
impeller, inches	54
H.P. cylinder, inches	13
L.P. cylinder, inches	25
Stroke, inches	12

MAIN CONDENSERS.

There is one Weir Uniflux condenser in the after part of each engine room, with a cooling surface of 14,007 square feet. The tubes have a pitch varying from $1\frac{1}{4}$ inches in the top to $\frac{21}{32}$-inch in the bottom. The exhaust trunk can be blanked off by wooden battens to permit a water pressure on the condenser for testing purposes.

The condensers are cross connected.

CONDENSER DATA.

Height, feet and inches	10–03
Width, feet and inches	9–09
Thickness of shell, inch	$00\frac{9}{16}$
Length between tube sheets, feet and inches	9–09½
Number of tubes	8,746
Diameter of tubes outside, inches	$00\frac{5}{8}$
Thickness of tubes, B.W.G	16

Cooling surface, square feet	14,007	
Exhaust nozzle, square feet	48	
Diameter of air pump suction, inches		14
auxiliary exhaust nozzle, inches		08
bleeder, inches		05
circulating water inlet and outlet, inches		29

FEED AND FILTER TANKS.

A feed and filter tank is located in the after outboard corner of each engine room on the working platform level. A horizontal partition divides the tank into two parts, the upper part being the filter tank.

The filter tank has an inner bottom of loose perforated plates, and is divided by vertical partitions into three chambers, which are filled with loofa from the inner bottom to near the top of the standing partitions and are covered with a galvanized woven-wire screen.

An 8-inch overflow pipe is fitted with a spring-relief valve, and leads from the top of the feed tank proper, and into the bilge through a funnel so arranged that any water passing through will be seen.

The two tanks are cross-connected and have the following additional connections: 10-inch main air-pump discharge; 5-inch auxiliary air-pump discharge; 3-inch vapor pipe; two 3-inch drains from the traps and feed-water heater.

Capacity of each feed tank, gallons	4,060
filter tank, gallons	687
Total capacity, gallons	4,747

MAIN FEED PUMPS.

There are four vertical, double-acting, simplex Blake pumps located in the engine rooms forward of the feed tanks on the working platform level. Pump size, $14\frac{1}{2}$ inches $\times$ $9\frac{1}{2}$ inches $\times$ 18 inches.

The pumps take their suction from either the main feed tank, reserve feed tanks or the air-pump channelways. The discharges from each pair of pumps unite in a $5\frac{1}{2}$-inch pipe and, after passing through a feed-water heater, combine into a

7-inch line in the after fireroom, which diminishes in size as it goes forward through the firerooms to 4 inches. Connections are fitted so that the feed water can by-pass the feed-water heater.

FEED-WATER HEATERS.

There is one direct-flow Wainright feed-water heater, located on the outboard fore-and-aft bulkhead in each engine room, forward of the main feed pumps, designed to maintain a feed temperature of 200 degrees at full power with a back pressure of 10 pounds. Connections are made to the auxiliary exhaust and to the auxiliary steam line, the pressure in the heater being controlled by a reducing valve.

DATA FOR ONE HEATER.

Diameter, inside, inches	32⅝
Length between tube sheets, feet and inches	5–10
Tubes, number	749
size, corrugated copper, inch	00¾
Heating surface, square feet	900
Diameter of feed inlet and outlet, inches	05½
auxiliary exhaust inlet, inches	08
drain, inches	03½

TURNING GEAR.

A twin cylinder 4-inch × 4-inch × 4-inch turning engine is bolted to the after bearing pedestal of each H.P. astern and L.P. turbine. This engine drives the turbine shaft through worm gearing, provision being made for disengaging it when not in use. In addition to the engine a hand-turning gear ratchet is provided.

Turbine-Lifting Gear.—Under the protective deck and rigidly bolted to the stringers are two parallel trolley tracks for each shaft, running the full length of the turbines, for lifting rotors and casings. The lifting gear consists of two 20,000-pound Harrington chain blocks with trolleys. The trolleys are fitted with sprocket gearing and locking gear so that it can be moved and locked in any position. Supporting rods are

bolted to the trolley path for suspending the upper casing and the rotor when making an examination of the lower casing.

FORCED-LUBRICATION SYSTEM.

The circulating pumps for the main, the auxiliary and the dynamo condensers are all housed and have independent systems of forced lubrication.

The turbine bearings, thrusts and the spring bearings are all connected to a forced-lubrication system in each engine room, consisting of two 10 × 9 × 12-inch vertical, double-acting, single Blake pumps, a gravity drain tank of 300 gallons capacity, and one oil cooler. The pumps and the drain tanks are cross-connected so that the system can be supplied by the pumps in either engine room.

The oil pumps take their suction from the drain tank and discharge through a quadruple-flow Alberger cooler and thence to the different bearings from which the oil drains by gravity to the drain tank.

DATA FOR ONE COOLER.

Diameter, in inches	24.67
Length between tube sheets, feet and inches	5-03
Number of tubes	417
Thickness of tubes, B.W.G	18
Cooling surface, square feet	365

ENGINE-ROOM AUXILIARIES.

In addition to the main condensers there is an auxiliary condenser of 722 square feet cooling surface in each engine room located inboard on the working platform, and connected through the auxiliary exhaust pipe to all the auxiliary machinery. Each condenser has an independent 5 × 6 engine and 10-inch circulating pump, and a Blake vertical simplex air pump.

Fire and Bilge Pumps.—Two Blake vertical simplex fire and bilge pumps 12-inch × 10-inch × 18-inch are located on the forward athwartship bulkhead in each engine room for pumping bilges, and with suction connection to the draining system

and sea and discharge to the firemain, sanitary system, pipe insulators and overboard.

Pipe Insulator Pumps.—In each engine room there is a 6-inch × 8-inch × 8-inch vertical simplex Blake pump with sea connections, for discharging cooling water around the main steam-pipe flanges at bulkheads in pipe passage to prevent transmission of heat through ships' structure to the magazines.

FIREROOM PUMPS.

In each fireroom there is installed one auxiliary feed pump and one fire and bilge pump. (For connections, see pump list.)

AIR COMPRESSOR PLANT.

Eight 11 × 11 × 12-inch Westinghouse steam-driven air compressors are located on the center line bulkhead, four in each engine room, together with four air reservoirs of 180,000 cubic inches capacity, for use in running pneumatic tools in the Engineering Department, cleaning boilers and for the gas-ejecting system for the guns.

Each compressor has a capacity of about 360 cubic feet of free air per minute at 150 pounds pressure.

A pneumatic main independent of the gas-ejecting system is led through the machinery space, with branches to the work-shop, evaporator and dynamo rooms, ice-machine rooms and the firerooms.

BOILERS.

There are twelve Babcock & Wilcox boilers, arranged in batteries of four, placed in three separate watertight compartments. The boilers are placed athwartship and have a common firing space of 12 feet 6 inches.

There are two smoke pipes, each 91.97 feet high, above the grates.

The boilers are fitted to burn coal and oil. Furnace and ashpit doors are inswinging, and are so balanced as to close automatically in case of a sudden outrush of steam. Between each of the furnace doors there is fitted a cast-iron frame

bolted to the furnace fronts and carrying two Babcock & Wilcox fuel-oil burners.

In these boilers the feed water enters the boiler and discharges upward against a mushroom cap in the steam drum, and, being deflected, falls into a scum pan, thus allowing what air is contained in the feed water to pass off into the steam drum.

DATA FOR BOILERS.

Number of boilers	12
Length of drum, feet and inches	19–00¾
Inside diameter of drum, inches	42
Thickness of drum plate, inch	00$\frac{11}{16}$
Total heating surface for one boiler, square feet	5,352.8
Grate surface for one boiler, square feet	119
H.S. for all boilers, square feet	64,234
G.S. for all boilers, square feet	1,428
Ratio of H.S. to G.S.	44.98
Number of boilers per smoke pipe	6
Height of smoke pipe above grate, feet and inches	91–11.6
Area through smokepipe, square feet	103.87
Ratio G.S. to smoke-pipe area	6.87
Size of generating tubes, inches	2 and 4
Total number of headers per boiler	31
Thickness of generating tubes, B.W.G.	8
4-inch generating tubes, B.W.G.	6
Exposed length of generating tubes, feet	8
Number of 2-inch generating tubes per boiler	1,100
4-inch generating tubes per boiler	31
4-inch circulating tubes per boiler	31
Inclination of tubes, degrees	15
Height of boiler, feet and inches	13–10$\frac{3}{8}$
Length of boiler, feet and inches	12–03¾
Width of boiler, feet and inches	20–00¾
Weight of one boiler complete, pounds	108,400
water in one boiler, middle gage cock, temperature 67.7 degrees, pounds	20,781
one boiler complete, water to middle gage cock, temperature of water 67.7 degrees F., pounds	129,181
Designed working pressure, pounds	215

MAIN AND AUXILIARY STEAM LINES.

The main steam pipes are arranged as shown in Fig. 1.

The main steam piping in the forward firerooms is 7½ inches in diameter, increasing in the middle firerooms to 9½ inches,

to 12½ inches in the after fireroom and 14½ inches in the pipe passages, decreasing to 14 inches in the engine rooms. Branches to boilers are 5½ inches.

There is no auxiliary steam line in the firerooms, but the main steam line has cross connections in each fireroom from which branches are taken for the auxiliary machinery therein. In the forward fireroom there is a 7½-inch cross connection with valves and castings, from which branches lead to the auxiliaries in the forward part of the ship. Cut-out valves are installed in each fireroom and in the cross connection so that any section can be cut out in case of accident.

An 8-inch cross connection extends across the forward part of the engine rooms joining the steam pipes from the pipe passage. From this cross connection two 7½-inch auxiliary steam pipes run aft, one in each engine room near the center-line bulkhead to the after part of the engine rooms where they join together to form a complete circuit. From these two branches steam lines lead to all the auxiliary machinery in the engine room and to the steering engine.

All valves in the steam line are fitted with by-pass valves.

AUXILIARY EXHAUST LINE.

The auxiliary exhaust lines lead from the forward fireroom aft on each side, through the pipe passages to the engine rooms, in the after part of which they unite. Branches and connection are provided to direct the exhaust steam into the feed-water heaters, the main condensers, the auxiliary condensers, the H.P. turbine and the L.P. turbine. In the middle fireroom there is a branch connection to atmospheric exhaust.

FORCED-DRAFT BLOWERS.

There are six hanging blower rooms built in the boiler rooms and suspended from the protective deck. In each of these blower rooms are installed two Sturtevant Multivane Centrifugal, cased, double inlet type fans. The impellers are 29¼

inches outside diameter and running at 965 revolutions per minute are capable of maintaining sufficient air for the maximum rate of combustion.

Each blower is independently driven by a G. E. Co. 45 H.P. electric motor direct coupled to the blower, with controlling apparatus installed so that they can be automatically started, stopped or adjusted for speed from the firerooms.

ASH HOISTS AND EXPELLERS.

The port ventilator in each fireroom is fitted with gear for hoisting ashes. A two-cylinder 4½-inch × 4½-inch ash hoisting engine is secured to the bulkhead in the uptake space for each boiler compartment, and is fitted with necessary sheaves, whips and appliances for handling ash buckets. There is installed in each fireroom a Metten Ash Expeller, discharging through the bottom of the vessel, operated by an 8-inch horizontal, single stage, centrifugal, DeLaval Pump driven by a DeLaval gearless steam turbine. A full description of these expellers will be found in Vol. XXIII, No. 4 of the JOURNAL OF THE A. S. OF N. E., November, 1911.

FUEL-OIL SYSTEM.

The ship is fitted with the Babcock & Wilcox mechanical atomizing system for burning fuel oil as an auxiliary to coal.

The fuel oil is carried in six double bottoms, C-86 to C-93 inclusive, with a capacity of 457.2 tons. For filling the fuel-oil double bottoms two 6-inch pipes are led across the ship under the main deck, with an 8-inch connection to a manifold in each engine room. Connections are fitted in the ship's sides for 6-inch hose. The individual compartments are filled through the manifold and suction pipes. An automatic float valve is fitted in each double bottom with electrical attachment for indicating when bottoms are filled to 95 per cent. of their net capacity.

From each of the double bottoms a 3.44-inch suction line

leads to the manifold in the engine room. All oil pipes are seamless-drawn steel with steel flanges screwed on and pipes expanded. The manifolds in the engine room are cross connected and fitted with gate valve which can be operated by extension rods in either engine room. Arrangements are made for pumping the oil overboard; the valves are fitted with locking gear so that only one discharge can be opened at a time. Two vertical duplex Blake Pumps 4-inch × 6-inch × 6-inch are installed, one in each engine room, and arranged so that through the cross connections of the manifolds either pump can be put on any bottom. The discharges from the two pumps unite into a common 3-inch discharge in the port engine room, which passes through the provision storerooms to the after fireroom, thence up and outboard and forward on the port side. In each boiler compartment there is located a pressure oil heater of 23.29 square feet of heating surface, U-shaped tube type. Branch connections are taken from the main supply line and so arranged that the oil can by-pass the heater. Steam for heating the oil is taken from the main steam line and trapped to drain line. A small pet cock is installed in the trap connections for testing for oil leaks in the heater. The supply pipes from heaters to boilers are led down and across boiler fronts and are fitted with valves controlled either in the firerooms or from deck.

The service pipe from each burner is $\frac{3}{8}$-inch diameter and discharge opening in the tip $\frac{1}{8}$-inch, the design being such that the oil is discharged as a revolving spray.

Air is admitted through non-return swinging doors in the outer plate of the air-box casing. The openings on the front of the air box are concentric with the tip of the burner and fitted with a thin metal disc provided with blades so arranged as to give the air a revolving motion in the same direction as the oil spray. During all the trials with coal, the burners were in place, being protected from the radiant heat of the fires by cast-iron shutters. When the oil trials took place, all that was necessary was to lower the shutters, put pressure on oil line

and open valves in burners. Upon completion of trial, shutters were again raised to position, and coal firing continued without the removal of burners. This system, therefore, avoids the objectionable feature of removing the burners when using coal.

ELECTRIC PLANT.

There are two dynamo rooms below the protective deck, one forward of the firerooms and the other abaft the after fireroom, each of which contains two 300-k.w. turbo-generators, built by the General Electric Company. They are of the 6-pole, compound-wound type, each capable of developing 2,400 ampères at 125 volts with a speed of 1,500 revolutions per minute. They are driven by direct-connected two-stage Curtis turbines with three rows of buckets in each stage, designed to operate at 200 pounds steam pressure and 28 inches of vacuum.

The turbine will carry its full rated load when operating non-condensing with normal steam pressure or with 20 per cent. below normal with 28 inches of vacuum. Each turbine is fitted with a governor to regulate its speed within 2 per cent. from no load to full load.

The valve mechanism is of the crosshead mechanical type, consisting of six valves operated by an oscillating steam lever driven from the main turbine shaft through a double reduction gear, and controlled by the inertia of the governor mounted on the primary-gear shaft. An emergency governor is also furnished, which closes the throttle valve in case the pressure at the exhaust nozzle rises above a predetermined point, or in case of an increase of speed of more than 8 per cent. above normal.

Each dynamo room has its independent condensing plant, consisting of a return-flow horizontal condenser, diameter 4 feet and $\frac{1}{2}$ inch and 7 feet between tube sheets. Each condenser has 2,030 tubes, $\frac{5}{8}$-inch outside diameter, No. 16 B.W.G., giving a cooling surface of 1,813.5 square feet. There is an independent air pump for each condenser, discharging

into a hot-well tank, from which the water is pumped to the main feed tanks. The centrifugal circulating pump is driven by a 6-inch × 6-inch vertical steam engine.

There are sixteen 36-inch searchlights mounted on the military masts and the boat-crane tops.

The lighting system consists of 2,500 lighting fixtures, not including special outlets for signal lanterns, truck lights, Ardois sets and special fixtures.

EVAPORATING AND DISTILLING PLANT.

The evaporator room is located on the berth deck just forward of No. 3 barbette. The distillers and the evaporator feed-water heaters are in the evaporator room trunks.

There are four evaporators and four distillers, Bureau design, with their accessories, two arranged for single effect and two arranged for double effect. The designed capacity of the plant is 25,000 gallons of potable water per twenty-four hours.

EVAPORATOR DATA.

Diameter, inside, feet and inches	6–01
Length over heads, feet and inches	5–08⅜
Tubes, number	120
diameter, inches	1½
Heating surface, square feet	312.9
Capacity, gallons per 24 hours	6,250
Diameter of steam connections (2), inches	2
vapor nozzle, inches	3
feed nozzle, inches	1½
blow valve, inches	2

DISTILLER DATA.

Diameter, inside, inches	15½
Length between tube sheets, feet and inches	4–08¼
Tubes, number	163
diameter, inch	00⅝
thickness, B.W.G.	16
Cooling surface, square feet	125.3
Diameter of circulating-water inlet and outlet, inches	3½
vapor inlet, inches	3
drain, inches	1½

EVAPORATOR FEED-WATER HEATER.

Diameter, inches	12⅞
Length between tube sheets, feet and inches	3–00⅞
Tubes, number	118
size, inch	00⅝
thickness, B.W.G.	16
Heating surface, square feet	59.4
Diameter of feed inlet and outlet, inches	2
vapor inlet and outlet, inches	4½

REFRIGERATING PLANT AND MAGAZINE-COOLING SYSTEM.

Five 3-ton Allen dense air ice machines are installed, three located aft for cold-storage service, ice making, scuttle butt and for after magazine-cooling system, one located amidship, for magazine cooling and scuttle butt, and one located forward for the forward magazine cooling. The after machines are piped so that any machine can be placed in the cold-storage rooms, scuttle butts, ice box or magazine-cooling system. The machines are so connected that any machine can be used for cooling any magazine.

Magazine Cooling.—In order to obtain a uniform temperature in the magazines, means are provided for supplying artificially-cooled air. The system is divided into three sections: one forward, one for amidship, and one for the after magazines, each with its separate cooler, blower fan and ventilating ducts. The cold air from the ice machines is led to an air cooler of condenser type, the cold air passing around the tubes. The cooled air is discharged through separate ducts to the magazines. The temperature of the delivered air is controlled by a thermostat installed in the ventilating air duct which, through a motor relay operating a small motor, controls the opening of the valve in the dense-air supply line.

In each magazine there is installed a thermostat which controls a magnetically-operated damper in the corresponding chilled-air duct. In the event of all the magazines of the group reaching the desired temperature. the dense-air valve is closed. On demand of any one magazine for refrigeration, the dense-air valve is opened to the necessary degree. A sig-

nal lamp is also fitted to indicate when any of the magazines of the group requires refrigeration.

COALING ENGINE.

There are two double-cylinder non-reversing steam-driven vertical 9-inch × 9-inch engines located in the coaling-engine room on the gun deck between frames 60–62. They are designed for a working pressure of 150 pounds, but are capable of withstanding the full boiler pressure.

Each engine is connected through a clutch and a 4-inch vertical shaft and bevel gear to a cross shaft under the main deck. The cross shaft is in two sections, and is fitted with a clutch at the center line of the ship, so that either side can be disconnected.

The 4-inch cross shaft drives a 4-inch shaft on both sides of the ship through bevel gearing. Ten vertical gypsies, five on each side of the main deck, connect with the fore-and-aft shafts through bevel gearing on the end of the gypsy shafts and beveled pinions fitted with sliding clutches and brakes on the fore-and-aft shafts, which are operated by bell cranks connected to tread plungers on the main deck. The clutch is normally held in mesh by a spiral spring around the shaft. Any one or all the gypsies may be disconnected and either side may be operated with either or both of the engines.

ANCHOR WINDLASS.

The anchor engine is of the horizontal double-engine type, diameter of cylinders 16 inches, stroke 16 inches, manufactured by the American Ship Windlass Co. It is designed for a working pressure of 150 pounds per square inch, but will safely carry the full boiler pressure of 210 pounds.

The reversing of the engine is done through a Stephenson link gear actuated by a hand wheel and screw so designed that one man can reverse the engine when it is working under full steam pressure.

The windlass has two vertical shafts driven by worm gear-

ing direct by the worms on the worm shaft. Each vertical shaft has on its upper end above the forecastle deck a wild cat, fitted with a locking head and ring so arranged that they can be operated together or independently of each other. Cast integral with the wild cat and on its lower part is a recessed ring to receive the brake band.

STEERING ENGINE.

The steering engine is of the vertical, two-cylinder type, Williamson Bros. Co., operated by automatic follow-up valves, so that the engine only functions when the steering wheels are turned. Diameter of cylinders, 21 inches; stroke, 16 inches. It is located in the steering-engine room on the starboard side of the fore-and-aft line, just abaft frame 105, access being from the starboard engine room. It is controlled from the steering stands on the conning-tower platform, in the conning tower, in the central station and in the steering-gear room aft, through wire-rope transmission; also by telemotor. The various controls are connected by clutches, so that any one may be operated independently of the others.

The steering-engine shaft is continued aft to the hand-steering room, where are located four large wheels for hand steering, provided with suitable clutches, so that the hand wheels may be disconnected when not in use. The main steering gear consists of a right-and-left-handed screw on which are two driving nuts direct connected by side rods to the crossheads on the rudder stock.

The tiller room is immediately abaft the hand-steering room.

MACHINE SHOP.

The machine shop is located amidship on the berth deck between the two engine-room hatches and aft of the evaporator room. It is reached from the gun deck and through the port engine-room hatch. The following is a list of the machines installed, and each is driven by an independent, enclosed electric motor:

1 28-inch by 46-inch by 10-foot extension-gap lathe;

1 12-inch lathe, taking 18 to 24 inches between centers, mounted on bed of the above and readily removable therefrom;

1 14-inch screw-cutting lathe, 4 feet between centers;

1 15-inch column tool-room shaper;

1 28-inch upright drill;

1 16-inch sensitive drill;

1 universal milling machine;

1 12-inch double emery grinder;

1 portable cylinder-boring machine;

6 machinist vises.

The tools are provided with the most modern attachments, including scroll and drill chuck, index head, automatic cross feed, swivel table, pipe vises, etc., and all necessary tools, drills and cutters.

FIRE MAIN.

A 6-inch fire main of lead-lined steel tubing, runs fore and aft under the protective deck, port and starboard, from between frames 28 and 39 to between frames 104 and 105. At these points they cross connect, forming a circuit. Between frames 90 and 91 there is another cross connection, to which the fire and bilge pumps in the engine rooms connect, through 5-inch risers fitted with a 5-inch angle stop valve in each engine room, operated in the engine rooms only. The fire and bilge pump in each boiler room is connected with the fire main through a riser, which branches and leads to starboard and port, connecting with the main, and fitted with 5-inch globe, stop, cut-out valves operated at the valves only. The valves are located in the boiler rooms between frames 74–75, 59–60 and 64–65, starboard and port. From frame 39 on the starboard side just under the protective deck, a 5-inch branch extends forward to frame 21, reducing to 4-inch. From frame 105 on the starboard side just under the protective deck a 5-inch branch extends aft to frame 125, reducing to 3½ inches. These branches have 5-inch globe cut-out valves, operated at valve and deck plates, forward, between frames 104–105, compartment C–109.

Starboard, between frames 124–125, a riser from the after fire-main system connects with the after sanitary system, having a 3½-inch globe stop-valve by-pass. There are two 6-inch globe stop valves in the forward cross connection on the starboard side, operated at deck plates in protective deck between frames 40–41, compartment 119. There are two 6-inch globe stop valves in the cross connection aft to bulkhead 90, operated at deck plates in protective deck between frames 90–91, compartments 106 starboard and port. The after cross connection is fitted with two 6-inch globe stop valves between frames 102–103 starboard and 103–104 port, compartments 109–110 ice-machine rooms. These cross-connection valves are for cutting out either the starboard or port side of the fire main proper. The fire main has three connections with the flushing system through 6-inch risers fitted with 6-inch by-pass stop valves operated in deck plates on protective deck between frames 39–40 in compartment A–119 starboard, 71–72 compartment C–106 port, and between frames 104–105 in compartment D–110 port.

FIRE-PLUG CUTOUT VALVES.

Two 2½-inch relief valves, set at 100 pounds, are fitted in the fire main, one between frames 42–43 and discharges into scupper between frames 47–48, and one between frames 117–118, discharging overboard at frame 116; both are located on starboard side of gun deck. All plug couplings are for 2½-inch hose. All fire plugs above the berth deck fitted about 13 inches above deck. All hose valves are gate valves. All other valves are globe valves.

FLUSHING SYSTEMS.

In the forward system a 6-inch flushing main runs forward under the port side of the gun deck from bulkhead 105 to just forward of frame 41. From this point are two branches: a 6-inch cross connection which connects with the riser from the fire main on the starboard side between frames 40–41; the other branch, 3½-inch, leads forward through the diagonal bulkhead between frames 39–40 to between 5–6, where it

branches to ward-room and junior officers' water closets on the gun and half deck; and at frame 20 a 3-inch branch leads to the ice machine. From the flushing main between frames 60–61 a branch supplies the fireman's wash room, admiral's and captain's bath rooms. At bulkhead 72½ the flushing main is connected with the fire main through 6-inch riser fitted with globe stop by-pass valve, and from this point branches lead to and supply the galleys and dynamo water service. Branches lead from the main between frames 82–82 to ice machine, warrant officers' water closet and chief petty officers' mess room. The main is fitted at frames 89–90 with a branch having a 2½-inch angle-key stop valve for flushing the oil system. Between frames 91–92 branches lead to general mess pantry. Between frames 98–99 a branch leads to chief petty officers' water closet and 3-inch relief valve set at 35 pounds. Between frames 101–102 a branch to ice machines, and sick quarters. Between frames 104–105 a branch leads to the laundry. At bulkhead 105 this system is connected with the fire main through 6-inch riser having an angle 6-inch by-pass valve. The flushing system is supplied through risers direct from the fire and bilge pumps in each fireroom and engine room.

An independent flushing system for the crew's water closet and washroom, located on the gun deck aft, is supplied by two motor-driven centrifugal pumps (direct connected), each delivering 500 gallons per minute. The pumps are located on the lower platform in compartments D–26–S. and D–26–P., between frames 124–125, and have direct connection with sea valves located between frames 126–127, and these discharges connect in a Y-casting from which a single pipe leads to the crew's water closet and wash room on the gun deck. This independent system has a by-pass connection with the fire main located on the protective deck between frames 124–125.

MATERIAL OF FLUSHING SYSTEM.

Pipes 1½-inch diameter and above are lead-lined steel tubing. Pipes 1¼-inch diameter and below are copper I.P.S. Relief valves are set at 35 pounds.

The system is tested at 100 pounds water pressure per square inch with the fixtures in the various rooms cut off, and to 40 pounds with fixtures under pressure, except between by-passes.

FRESH-WATER SYSTEM.

The main fresh-water tanks in compartments A–4, A–5, A–6, A–7 and A–8, located in the hold between frames 10 and 18, are supplied by 2½-inch filling plugs located between frames 15–16 starboard and port. The 2½-inch distiller main runs from the evaporators along the port side of protective deck parallel with the flushing main to the main tanks.

From the pump manifolds a 1½-inch pipe leads aft on port side of protective deck to frame 47 and up to gravity tank on chart house.

A 3-inch distributing pipe leads aft on the starboard side of the protective deck to frames 90–91, and supplies the various tanks and connects with the reserve feed-water pipe through by-pass at frames 56–57.

Main gravity tanks are filled by fresh-water pumps 1 and 2, located in pump room A–53, through fresh-water mains from main tanks, and can also be filled from filling plugs at ship's sides.

There are two electrically-operated fresh-water pumps located in the pump room, A–53 upper-platform deck between frames 11½–14, starboard side. Each pump is a direct-acting triplex plunger pump, with designed capacity of 60 gallons per minute against the working head, operated by a 3-horsepower motor. The plungers have a diameter of 4 inches with 6-inch stroke; size of piping, 2½-inch.

MAIN DRAIN.

A main drain, 15½-inch diameter throughout, extends from the after end of the forward fireroom, about 6½ inches from and parallel to the starboard coal-bunker bulkhead to the for-

ward end of the starboard engine room, where it branches, one branch passing athwartship along bulkhead goes to the port engine room. From the athwartship branch two branches, one in the port and the other in the starboard engine rooms, extend aft, each branch connecting with the centrifugal pumps, and the overboard discharge is through the condenser and outboard-delivery pipe.

In each engine and fireroom is fitted a 15½-inch stop check, having a short neck, and placed with the opening about 3¾ inches above the inner bottom.

The valves are operated in the compartment they drain by a hand wheel about 30 inches above the floor and in the deck plates at the protective deck. No other valves are fitted to the main drain proper. As auxiliary connections to the main drain proper there are connections near each fire and bilge pump in the engine rooms.

SECONDARY DRAIN.

The forward secondary drain extends from a manifold at bulkhead 18 into and through the boiler-room spaces. This drain is 4⅝-inch inside diameter forward of the fire and bilge pumps in No. 1 boiler room, and from this pump to the bilge well of No. 1 boiler room it is 5 inches inside diameter, and there to the after end of No. 3 boiler room is 5½ inches inside diameter.

The after secondary extends from the after trimming tank into and through the starboard engine room. This drain is 5½ inches inside diameter through the engine room, and reduces to 4⅝ inches and 3 inches as it goes aft. A secondary drain also extends through the port engine room. These secondary drains are connected to each fire and bilge pump the full diameter of their respective pump suctions; stop valves are fitted in these connections. A Macomb strainer is fitted to each of these main lines—one in the forward boiler room, and one in each engine room. Branches of the same diameter as

the secondary drain are led to bilge wells in the after end of each boiler room and the forward and after end of each engine room.

These branches dip into the wells and are fitted with Macomb strainers at the wells and stop check valves between the wells and the secondary drain.

Two 6-inch hand pumps are furnished, for which there are connections at each fire and bilge-pump manifold. The chain locker drains into the drainage tank, which is connected to forward secondary drain.

From the secondary drains 3-inch branches are taken for draining all double-bottom compartments inboard of the outermost longitudinal below the armor shelf, except the reserve feed-water and fuel-oil tanks, and for draining the trimming tanks drainage tanks and dynamo condenser compartments. For double-bottom compartments in way of the boiler and machinery spaces the branches are fitted with stop-lift-check valves; forward and abaft the machinery spaces the branches are fitted with stop-check valves.

Valves in the secondary drains and in branches to the bilge wells and double-bottom compartments in way of the machinery spaces are operated 30 inches above the floor plates.

Valves controlling suctions forward and abaft the machinery spaces, except trimming tank suctions and valves in the shaft alleys, are operated at the valves and on berth or protective deck in deck plates. Valves located in shaft alleys are operated at the valves only.

Valves controlling forward trimming-tank suctions are operated from the berth deck, and those controlling after trimming-tank suctions, from the steering-gear plat.

AIR ESCAPES AND OVERFLOWS FROM COAL BUNKERS.

Seven-inch air escapes are located on the protective deck leading from the lower bunkers to about 5 feet above deck and with a return U to the firerooms.

From Compartment.	To Compartment.	Location — Deck.	Location — Bet. frames.
B-4 S.	B-1	Protective	47-48
B-4 P.	B-1	Protective	47-48
B-6 S.	B-1	Protective	52-53
B-6 P.	B-1	Protective	52-53
B-8 S.	B-2	Protective	56-57
B-8 P.	B-2	Protective	56-57
B-10 S.	B-2	Protective	59-60
B-10 P.	B-2	Protective	59-60
B-12 S.	B-3	Protective	63-64
B-12 P.	B-3	Protective	63-64
B-14 S.	B-3	Protective	67-68
B-14 P.	B-3	Protective	67-68

FLOODING COAL BUNKERS.

The upper bunkers are flooded by inserting fire hose through scuttles. The lower bunkers are flooded from the fire main through 6-inch branches having globe stop valves operated at the valves and indicated deck plates in protective deck.

Compartment. flooded.	Deck plate in compartment.	Location — Deck.	Location — Bet. frames.
B-4 S.	B-101	Protective	47-48
B-4 P.	B-100	Protective	47-48
B-6 S.	B-101	Protective	52-53
B-6 P.	B-100	Protective	52-53
B-8 S.	B-101	Protective	54-55
B-8 P.	B-100	Protective	54-55
B-10 S.	B-103	Protective	61-62
B-10 P.	B-102	Protective	61-62
B-12 S.	B-103	Protective	62-63
B-12 P.	B-102	Protective	62-63
B-14 S.	B-103	Protective	67-68
B-14 P.	B-102	Protective	67-68

TRIMMING–TANK CAPACITY.

Compt.	Location.	Salt water, tons.
A-1	Stern to frame 5, keel to upper platform deck,	97.6
A-2	Frame 5 to frame 10, keel to upper platform deck,	158.3
D-8	Frame 122 to frame 127, keel to steering-gear flat,	116.1
D-9	Frame 127 to stern, keel to steering-gear flat,	45.4

COAL BUNKER CAPACITIES.

Capacities are calculated at 43 cubic feet to the ton and to 6 inches below the bottom of beams.

Lower Bunkers.

Compt.		Location.	Cubic feet.	Tons.
B-4	S.	Frames 45 to 49	3,517.49	81.80
B-4	P.	Frames 45 to 49	3,555.51	82.69
B-6	S.	Frames 49 to 54	5,888.70	136.95
B-6	P.	Frames 49 to 54	5,588.51	136.94
B-8	S.	Frames 54 to 58	6,327.09	147.14
B-8	P.	Frames 54 to 58	6,322.80	147.14
B-10	S.	Frames 58 to 62	5,876.17	136.66
B-10	P.	Frames 58 to 62	5,876.27	136.66
B-12	S.	Frames 62 to 66	5,886.77	136.90
B-12	P.	Frames 62 to 66	5,887.03	136.91
B-14	S.	Frames 66 to 70	4,841.19	112.59
B-14	P.	Frames 66 to 70	4,844.09	112.65
				1,504.03

Upper Bunkers, Berth Deck.

B-104	P.	Frames 45 to 48	2,681.23	62.35
B-105	S.	Frames 45 to 48	2,687.84	62.51
B-106	P.	Frames 48 to 52	3,865.24	89.89
B-107	S.	Frames 48 to 52	3,864.04	89.66
B-108	P.	Frames 52 to 56	3,853.27	89.61

Compt.	Location.	Cubic feet.	Tons.
B-109	S. Frames 52 to 56	3,853.27	89.61
B-110	P. Frames 56 to 60	4,018.88	93.46
B-111	S. Frames 56 to 60	4,018.88	93.46
B-112	P. Frames 60 to 64	3,564.53	82.90
B-113	S. Frames 60 to 64	3,564.53	82.90
B-114	P. Frames 64 to 68	3,828.87	89.04
B-115	S. Frames 64 to 68	3,828.87	89.04
B-116	P. Frames 68 to 72	2,609.13	60.68
B-117	S. Frames 68 to 72	2,607.71	60.64
	Total, . . .		1,135.75
	Lower bunkers, .		1,504.03
	Total bunker capacity,		2,639.78

RESERVE FEED-WATER CAPACITIES, INNER BOTTOMS.

Compt.	Location.	Tons F. W.
B-86	P. Frames 45 to 49	51.4
B-87	S. Frames 45 to 49	51.4
B-88	P. Frames 49 to 54	65.9
B-89	S. Frames 49 to 54	65.9
B-90	P. Frames 54 to 56	26.6
B-91	S. Frames 54 to 56	26.6
	Total, . . .	287.8

DOUBLE-BOTTOM CAPACITIES, EXCLUSIVE OF RESERVE FEED-WATER AND FUEL-OIL TANKS.

Compt.	Location.	Tons F. W.
A-93	Centerline, 10-14	47.0
A-94	Centerline, 14-18	55.0
A-95	Centerline, 18-24	82.8
A-96	Centerline, 24-30	89.3
A-97	Centerline, 30-34	62.2
A-98	Centerline, 34-40	96.9
A-99	Centerline, 40-45	82.3
B-92	P. 56-58	26.6

DOUBLE-BOTTOM CAPACITIES (CONTINUED).

Compt.	Location.	Tons F. W.
B-93	S. 56-58	26.6
B-94	P. 58-62	53.0
B-95	S. 58-62	53.0
B-96	P. 62-66	53.0
B-97	S. 62-66	53.0
B-98	P. 66-70	53.0
B-99	S. 66-70	53.0
C-94	P. 90-95	65.0
C-95	S. 90-95	65.0
C-96	P. 95-100	63.5
C-97	S. 95-100	63.5
C-98	P. 100-105	58.6
C-99	S. 100-105	58.6
D-97	Centerline, 105-110½	84.3
D-98	Centerline, 110½-115	97.6
D-99	Centerline, 115-122	132.9
	Total, . . .	1,575.7

PUMPS AND PUMP CONNECTIONS.

No.	Pumps.	Size.	Kind.	Suction pipes from—		Discharge pipes to—		Location.
		inches.		*ins.*		*ins.*		
2	Main air	One 20-in., two 36-in. by 21.	Weir Dual twin vert. bucket, single acting.	14	Condenser	10	Feed tank	One each engine room.
				5	Turbine drains.			
2	Main circulating	12×22×11	Vert. compound, centrifugal.	2-21	Sea	29	Condenser	One each engine room.
				15½	Main drain	29	Overboard.	
4	Main feed	14½×9½×18	Vert. piston, double acting, single.	5	Feed-tank connection pipe	4	Main feed discharge	Two each engine room.
				5	Channel-way, cross-connection pipe.			
3	Auxiliary feed	14½×9½×18	do	5	Feed suction pipe	4	Auxiliary feed main	One each boiler room.
				4	C. and R. pipes from ship's sides (*a*)	3	Overboard.	
				5	Reserve feed tanks	3½	Reserve feed tanks (*a*)	
				2½	Hose connections.			
4	E. R. fire and bilge	12×10×18	do	5	Sea	5	Fire main	Two each engine room.
				5	E. R. Bilge	5	Overboard.	
				4½	Shaft alley.			
				5	C. and R. drainage manifold.	4½	Oil cooler.	
				1½	Hose Connection	2½	Hose connection.	
3	Fireroom fire and bilge	12×10×18	do	5	Fireroom floors	5	Fire main	One each boiler room.
				5	Sea	5½	Overboard.	
				5	C. and R. drainage manifold.	5	Sanitary.	
				1½	Hose connection	2½	Hose connection.	
2	Oil cooler, circulating	10×9×12	do	5	Sea	4½	Oil cooler	One each engine room.
1	Fresh water	6×6½×6	Vert. piston, double acting, single.	3	Reserve feed tanks	3	Feed tanks	One in port eng. room.
2	Dynamo turbine bearing, circulating.	5×3×4	Vert. piston, double acting, duplex.	2	Sea	1½	Dynamo turbine bearing	Dynamo room (one in each).
2	Fuel-oil supply	7½×4×8	do	3½	Storage tanks	3	Burners	One each engine room.
4	Forced-lubrication serv	10×9×12	do	5	Lubricating oil tanks	4½	To coolers	Two each engine room.
						3½	From coolers to bearing.	
2	Dynamo-room hot well	4½×5×6	do	2½	Dynamo-room hot-well tank	3	Feed tank	Dynamo room (one in each).
2	Distiller, circulating	12×14×16	Vert. piston, double acting, single.	6½	Sea	6	Distillers	Near evaporators.
						5	Fire main.	
2	Auxiliary air	7½×14×16	Vert. simplex piston.	5	Auxiliary condenser	4½	Main feed tanks	One each engine room.
2	Dynamo condenser, air	9×18×12	Vert. piston, double acting, single.	7	Dynamo condenser	6	Dynamo-room hot well	Dynamo room.
2	Dynamo condenser, circ		Centrifugal Vertical engine.	9	Sea	9	Dynamo condenser	Dynamo room.
2	Evaporator feed	4½×5×6	Vert. piston, light serv., dbl. act. single.	2	Sea	2	Evaporators	Near evaporators.
				2	Evaporator feed heaters.			
1	Distiller, fresh water	4½×5×6	Vert. piston, light service.	1½	Bottom of distillers	2½	Fresh-water tanks	Engine room.
				2½	Test tank	...	Main feed tanks.	
							Reserve feed-water tanks.	
2	Pipe insulator, circulating	6×8×8	Vert. piston, double acting, single.	3½	Sea	3	Pipe insulators	One each engine room.
3	Ash ejector		Centrifugal	5	Sea	4	Ash ejectors	One each fireroom.

(*a*) No. 2 boiler room only.

TABLE I.—STANDARDIZATION TRIAL DATA U. S. S. "WYOMING."

Rockland, Maine. July 19, 1912.

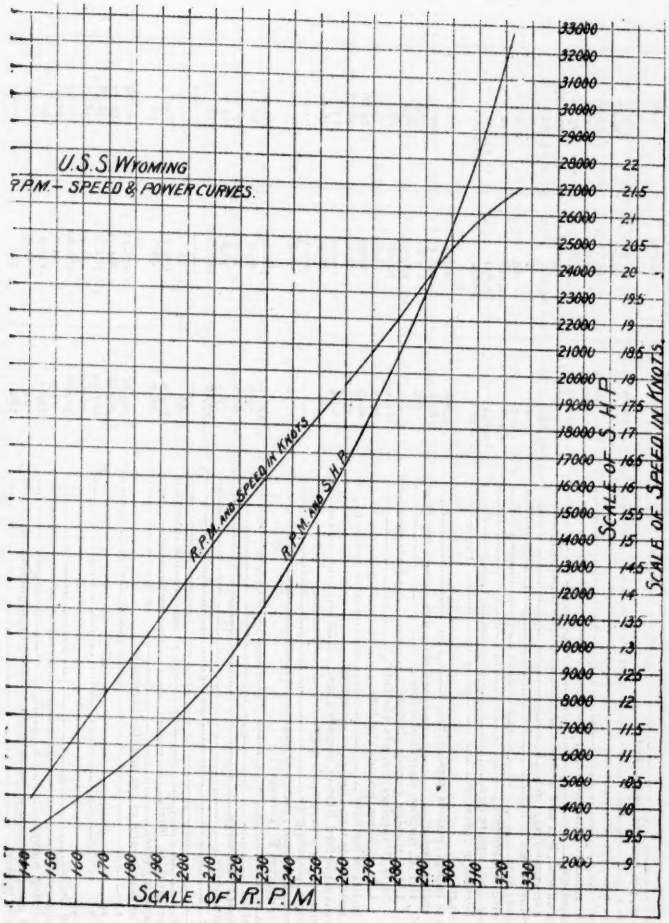
Number of run.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	17	18	19	21	22	23	24	25
Time on course, minutes... seconds...	5 41.83	5 51.45	5 56.8	4 40.85	4 44.3	4 42.3	3 59.4	3 57.9	4 0.4	3 29.5	3 23	3 24.63	3 9.13	3 13.1	3 7.67	2 52.1	2 55.23	2 49.67	2 44.73	2 53 37	2 43.3	2 55.37	2 43.23
Revolutions, to go 1 knot:																							
Shaft No. 1	894.2	801.2	815.1	786.1	803.5	790.7	827.3	819.8	821.7	840.4	832.3	840.7	843 9	874.7	847.1	893 7	914.7	888.6	903 5	930.9	885.4	938.7	879.1
No. 2	786.3	909.9	922.8	898.9	923.	902.3	962.2	953.1	958.4	970	956	969.5	913.2	943 9	920.2	869.6	886.6	868.4	878 8	907.9	859.9	913.1	853.5
No. 3	885.	890.6	918.9	891.4	918.1	888.7	813.9	801.1	809.9	868.1	871.3	858.3	871.6	876.8	865.1	885.2	897.8	875.8	889.1	925.4	883.5	938.2	879
No. 4	806.7	810.1	836.	796.4	820.5	795.8	835.8	826.2	834.5	834.3	834.7	837.6	854.4	883	858.9	883.8	901 7	874.5	893.7	925.8	882.9	938.5	879.3
Average	843.05	853	873.25	843.2	866.3	844.4	859.85	850.1	856 15	878.2	873.6	876.5	870.8	894.6	872.85	883.1	900.25	876.85	891.25	922.5	877.95	932.15	872 8
Revolutions, per minute:																							
Shaft No. 1	138.00	136.84	137.17	168.26	169.68	168.17	207.39	206.76	205.11	240.81	245.99	246.60	267.74	271.94	270.81	311.44	313.36	314.27	328.97	322.22	325.40	321.19	323.30
No. 2	156.95	155.41	155.28	192.40	194.91	191.92	241.21	240.37	239.23	277.93	282.53	284.38	289.71	293.46	294.18	303.03	304.32	307 13	319.96	314.27	316.03	312.44	313 90
No. 3	155.32	152.11	154.63	190.80	193 88	189 02	204.04	202.04	202.17	248.73	257 52	251.77	276.51	272.58	276.55	308.47	307.47	309.73	323.70	320.32	324.70	321.01	323.27
No. 4	141.58	138.36	140.68	170.45	173.26	169.26	209.53	208.38	208.30	239.06	246.69	245.69	271.05	274.53	274.57	308.00	308.79	309.29	325.40	320.45	324.48	321.13	323.39
Average	147.96	145.68	146.94	180.48	182.93	179.59	215.54	214.39	213.70	251.47	258.18	257.11	276.25	278.13	279.03	307.73	308.46	310 10	324.50	319.31	322.65	318.94	320.96
Speed in knots	10.53	10.24	10.09	12.82	12.66	12.75	15.04	15.132	14.98	17.18	17.73	17.59	19.04	18.64	19.18	20 92	20.54	21 22	21.85	20.77	22.05	20.53	22.06
S.H.P., Shaft No. 1	552	588	521	993	984	992	1,867	1,923	1,743	2,986	3,149	3,329	4,206	4,569	4,306	7,101	7,174	7,385	8,553	8,056	8,298	7,837	7,986
No. 2	989	979	978	1,866	1,852	1,823	2,787	3,894	3,588	5,892	5,990	6,029	6,171	6,691	6,237	6,333	6,817	7,033	8,223	7,857	7,774	7,624	7,691
No. 3	916	913	897	1,794	1,919	1,720	1,776	1,778	1,658	3,681	3,889	3,852	4,839	4,988	4,895	7,157	7,226	7,372	8,675	8,456	8,799	8,539	8,599
No. 4	524	526	521	989	953	948	1,781	1,875	1,771	2,725	2,985	3,145	4,201	4,694	4,448	6,684	6,546	6,588	7,777	7,306	7,593	7,322	7,664
Total, all shafts	2,981	3,006	2,917	5,642	5,708	5,483	8,211	9,470	8,760	15,284	16,013	16,355	19,417	20,632	19,886	27,275	27,763	28,377	33,228	31,675	32,464	31,322	31,940
Turbines used	H.P.C., I.P.C., M.H.P. and L.P.						I.P.C., M.H.P. and L.P.									M.H.P. and L.P.							

TABLE II.—U. S. S. "WYOMING."—TRIAL DATA.

	4-hour full-power trial.	24-hour 19-knot endurance trial.	22-hour 12-knot endurance trial.	2-hour coal and oil burning trial.

TABLE II.—U. S. S. "WYOMING."—TRIAL DATA.

.....	4-hour full-power trial.	24-hour 19-knot endurance trial.	22-hour 12-knot endurance trial.	2-hour coal and oil burning trial.
Date of trial	July 20, 1912.	July 20–21, 1912.	July 21–22, 1912.	July 22, 1912.
Speed in knots	21.223	19.209	12.266	20.979
Slip of propellers, per cent. of own speed:				
Propeller No. 1	18.89	13.9	5.52	17.17
No. 2	16.15	19.15	19.23	14.35
No. 3	17.22	15.63	23.51	16.91
No. 4	18.66	14.6	4.21	18.88
Mean, all propellers	17.73	15.82	13.12	16.83
Draught, mean on trial, feet and inches	27–06¾	27–05½	27–05½	27–06 1/16
Displacement corresponding, tons	25,085	24,980	25,004	25,008
Number of turbines used	4	5	6	4
boilers used	12	12	8	12
Grate surface used, square feet	1,428	1,428	952	1,428
Heating surface used, square feet	64,234	64,234	42,822	64,234
Number of oil burners in operation	None.	None.	None.	All.
Pressures (average):				
Main steam at boilers, pounds gage	208.6	206.9	206.6	216
engine rooms	180.8	192.4	201.2	.....
In steam belt, H.P.C. turbine, pounds absolute	5.9	2.6	135.8	.....
I.P.C. turbine, pounds absolute	.....	185.5	71.4	.....
M.H.P. turbines, pounds absolute	164.4	100.2	29.1	145
L.P. turbines, pounds absolute	15.7	11.2	3.7	.....
Barometer, inches of mercury	29.9	29.93	29.86	29.84
Vacuum in main condensers, inches of mercury	27.9	28.1	28.35	27.38
Auxiliary exhaust in feed heaters, pounds gage	13.4	.....	13	.....
Forced-lubrication system, pounds gage	20	18.5	16.6	.....
Fuel-oil system, pounds gage	.....	.....	.....	156.75
Air pressure in firerooms, inches of water	1.7	0.58	.....	1.86
Temperatures, degrees F. (average):				.....
Main steam	370.7	386.6	382.9	.....
injection	59.8	59.5	66.2	.....
overboard discharge	84.2	79.2	78.9	.....
Air-pump discharge	77.3	72.8	74.7	.....
Feed water	177.6	188.6	218	183.5
Fuel oil	.....	.....	.....	147.3
Outside air	66	95	.....	.....
Engine-room working level	105.9	109.1	104.7	.....
Fireroom working level	88.7	90.6	98	98
Revolutions or double strokes per minute (average):				
Shaft No. 1	323.88	276.15	160.68	313.48
No. 2	313.28	294.05	182.71	303.16
No. 3	317.33	281.80	198.48	312.50
No. 4	322.93	278.40	158.49	320.09
Mean of all shafts	319.36	282.60	175.09	312.31
Main air pumps	23.9	24.2	25.3	.....
circulating pumps	187.1	175.2	147	.....
feed pumps	24	18.7	15.5	.....
Forced-draft blowers	789	712	.....	.....
S.H.P. (average):				
Shaft No. 1	7,959	4,371	688	6,557
No. 2	7,944	6,553	1,706	6,857
No. 3	7,974	5,301	2,202	7,660
No. 4	7,560	4,559	757	7,815
Total, all shafts	31,437	20,784	5,353	28,889


U.S.S. WYOMING
9 P.M. – SPEED & POWER CURVES.
R.P.M. AND SPEED IN KNOTS
R.P.M. AND S.H.P.
SCALE OF S.H.P.
SCALE OF SPEED IN KNOTS.
SCALE OF R.P.M.
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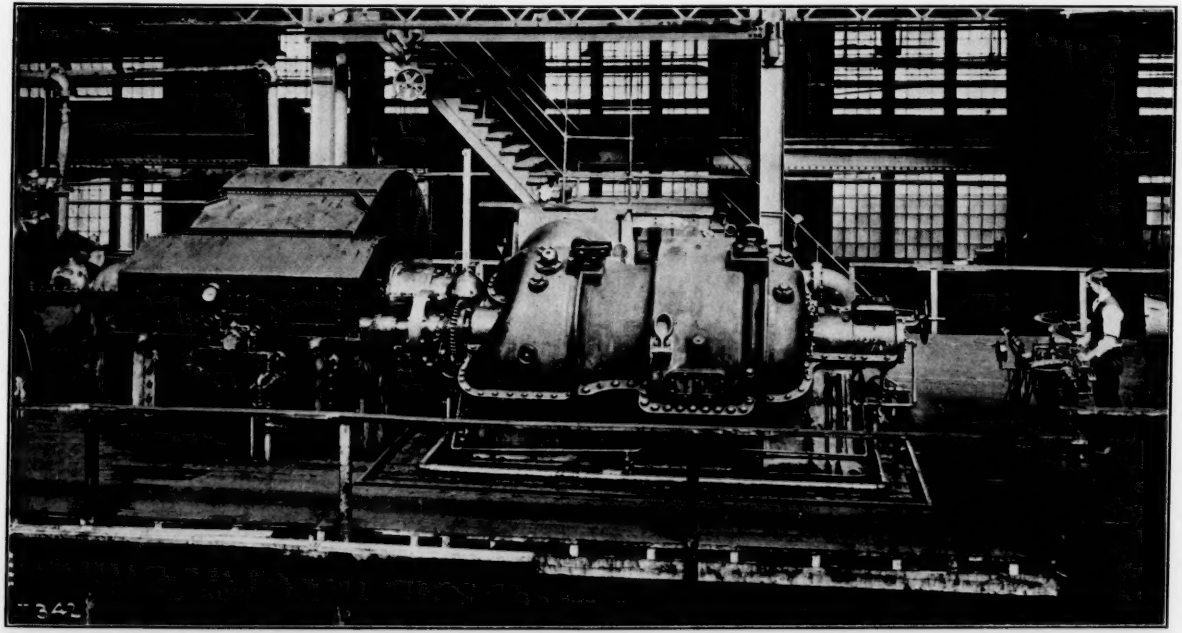


Plate I.—View of the Port Unit of the "Neptune" Showing the Turbine, Reduction Gear and Operating Stand.

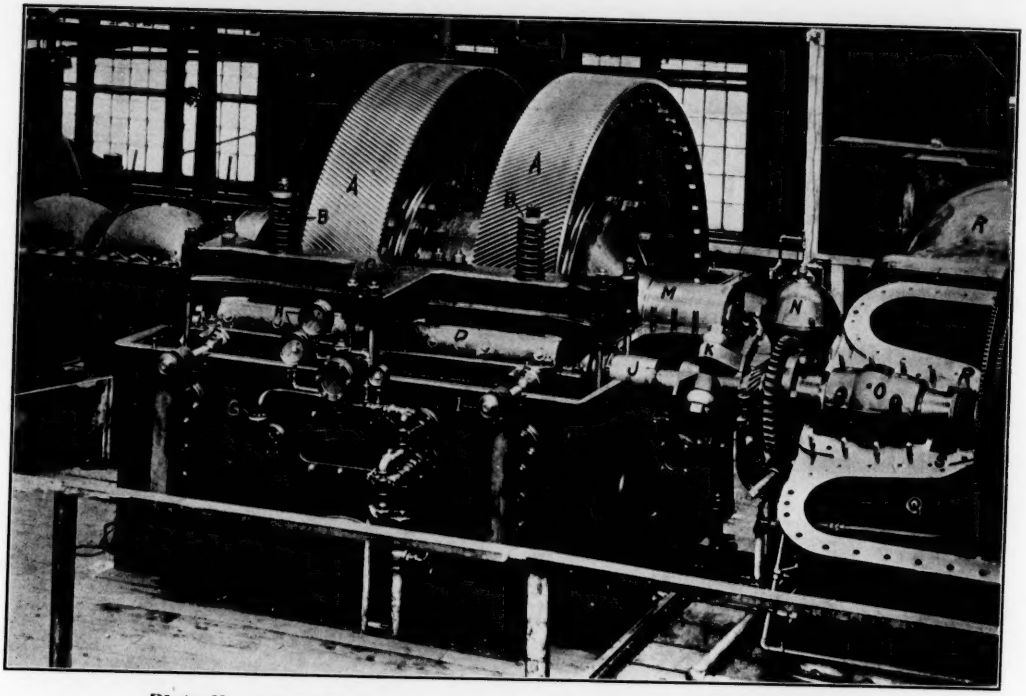


Plate II.—VIEW OF THE REDUCTION GEAR WITH THE COVER REMOVED.

A are the gear wheels showing the helical teeth. B the balance springs. C the upper bed of the floating frame. D the floating frame. E the struts which keep the gears in the correct depth of mesh. F the control valve of the floating frame. G the drain from the control valve. H the ahead and astern dynamometer gages. I, strut-adjusting screws. J, end of pinion. K, flexible coupling (old type). M, gear journal. N, turbine governor, and below governor drive and turning gears O, turbine bearing. P, gland-packing rings. Q, exhaust chamber of turbine. R, exhaust pipe connected to lower part of turbine. S, low-pressure blading of turbine. T, counter-gear driving gears.

NOTES ON THE PERFORMANCE OF THE REDUCTION GEAR AND OF THE BRIDGE-CONTROL SYSTEM OF THE U. S. S. *NEPTUNE*.

By Lieutenant W. W. Smith, U. S. Navy, Member.*

The United States naval collier *Neptune* is equipped with high-speed turbines which drive the propeller shafts through Melville and Macalpine reduction gears. The purpose of the reduction gear, interposed between the turbine and propeller, is to reconcile the high speed necessary for good turbine economy with the low speed required for good propeller efficiency, so that each will run at the speed for which it is best suited; thereby giving the maximum over-all economy for the propelling machinery. In this way the high economy of the high-speed turbine can be combined with the high efficiency of a low-speed propeller to give a very high over-all economy. The turbine shaft of high rotative speed is connected to the pinion, and the propeller shaft of low rotative speed is connected to the gear; the ratio of reduction being 9.04 to 1. At full speed the speed of the turbine is 1,220 r.p.m., and that of the propeller 135.

The reduction gear is of the double-helical type with involute teeth inclined at an angle of 30 degrees to the axes, and is enclosed in an oiltight casing, the lower part of which forms the bed plate, in which are mounted the gear in rigid bearings, and the floating frame, which carries the pinion on flexible supports. The axis of the pinion is held in a vertical plane parallel to that of the axis of the gear by transverse struts,

* Lieut. Smith served as Inspector of Machinery at the works of the Westinghouse Machine Company, and at the shipbuilding plant of the Maryland Steel Company, during the construction and installation of the reduction-gear machinery of the *Neptune*, and on board that vessel during the seven and a-half months that she has been in service carrying coal to the Fleet. In this time she has steamed 5,000 miles.

which stand between the floating frame and the bed plates. The pinion, which is hollow, is flexibly connected to the turbine shaft by a flexible shaft extending through it to the end distant from the coupling, where it is secured, and a coupling flexible to longitudinal movement. There is an abundant forced supply of oil which serves both as a lubricant and as a cooling medium.

THE WESTINGHOUSE HYDRAULIC FLOATING FRAME.

(See Plates III and IV.)

The purpose and principle of action of the Westinghouse hydraulic floating frame are to automatically align the pinion so as to distribute the pressure uniformly over the entire length of tooth contact.

The pinion is carried in a heavy cast-iron frame which is supported by pistons, both on the bottom, for ahead rotation, and on the top, for astern rotation. The floating frame is held in the central position between the upper and lower beds by automatic-control valves, which admit or exhaust oil from the cylinders as required.

The floating frame is amply rigid to withstand deflection. It is divided horizontally and bolted together with fitted bolts. The pinion is rigidly supported by three bearings, which consist of brass bushes lined with white metal. The bearings are securely held in the floating frame, which is bored to receive them. In the bearing seat a groove is turned so as to form an oil channel when the bearing is in place. This channel connects with the longitudinal supply channel in the floating frame and with the longitudinal supply channels on the inside of the bearing. In addition to the outlets to the bearings the supply channel is tapped by six $\frac{1}{4}$-inch square holes abreast each pinion. The supply channel connects with a brass pipe under the floating frame which leads from the lubrication system. To provide for the movements of the floating frame there is a flexible section with a ball-and-socket joint at each end. These joints are fitted with packing rings to prevent leakage.

The two pinion sections are entirely enclosed by the pinion boxes, except where they mesh with the gear, and at these points the floating frame extends in close to the gear so as to guide the oil into the meshing teeth.

Directly above and below each bearing there is a shallow cylinder, in which there is a piston composed of two steel plates held together by screws. The piston is packed with a cup leather and a bronze packing ring, the latter being provided to retain the oil in case the former fails. Behind the cup leather there is a steel-spring ring which holds it out against the wall of the cylinder. The pistons are proportioned so that each will exert a force equal to the load on its bearing. As the load on the middle bearing is approximately twice that of an end bearing, the area of the middle cylinder is about twice that of an end one.

The bottom cylinders connect with a longitudinal channel, so that the three cylinders are always in communication. This channel connects with one that leads out through the radius arm and connects with the bottom-control valve. The top cylinders are also in communication, and connect with the top-control valve.

In the bottom and also the top of each bearing there is a short longitudinal collecting groove, from which a hole leads down into the cylinder beneath; also from the latter up into the cylinder above. In the cylinder there is a check valve which prevents the oil from escaping from the cylinder when the bearing is not pumping, as in starting, etc.

Below and also above the floating frame there is a bed having three piston seats on which the pistons rest. The middle seat is slightly higher than the two end ones, so that, in case the oil supply to the cylinders fails, the floating frame will rock or tilt mechanically on the middle seat as necessary for its compensating action, thus preventing heavy local pressure which would damage the teeth. The lower bed is bolted to the gear box, and the upper one is also secured to the gear box by turned-steel columns. To prevent the floating frame from moving fore-and-aft with the pinion there is a stop lug on

each side of the middle lower cylinder. The total clearance between the floating frame and the middle seats is .170 inch, and between the end seats .113 inch.

On the outboard side of the floating frame and in the middle there is a bracket which carries the upper bearing of the suspension link. At the lower end of the suspension link there is also a bearing carried in a bracket bolted to the bed plate. The suspension link holds the bracket bearing in a horizontal plane passing through the center line of the gears, so that if the floating frame is raised or lowered it will hinge about the bearing as a center. The purpose of the suspension is to provide a center, so that the movements of the radius arm will correspond exactly with those of the floating frame, and, at the same time, to multiply the latter for more sensitive control.

As the floating frame is supported by the pistons directly beneath the bearings, and as it is held against horizontal movement by the struts, there is practically no load on the suspension link, which merely acts as a center for the hinging movement of the radius arm. Without this suspension the floating frame would cant, and the radius arm would not move correspondingly, so that the control valves could not maintain it in the mid position.

To the middle of the floating frame on the inboard side a brass radius arm is bolted. In it there are two drilled channels which at one end connect with the upper and lower longitudinal channels leading to the cylinders of the floating frame, and at the other with the upper and lower control valves. Under the control valves there are raised seats which are accurately ground to prevent leakage. Near the middle of the arm there is a piston with two packing rings, which rest against the bored inside of the valve body, to prevent leakage from the exhaust chamber of the valve into the gear box in case water is used to support the floating frame.

The body of the control valve, which is of brass, is bolted to the lower part of the gear box and encloses the end of the radius arm in the exhaust chamber of the valves. The exhaust chamber has a drain leading into the gear box, and a cover

on the outer end. The body contains the two control valves which are located above and below the radius arm. The exhaust ring of the control valve is held against a shoulder in the body by a helical steel spring which is held in compression by the supply-valve seat. The latter is forced into the body and acts both as a seat and a guide for the supply valve, which is held on its seat by a helical spring which is held in compression by the cap. In the supply-valve stem, which extends down slightly below the exhaust ring, there is an axial hole which is connected by radial holes at the top and bottom with the annular space below the valve and the supply chamber respectively. Both inlet chambers are connected with the hydraulic oil system, and both valve chambers to the dynamometer gages.

The point of contact with the seat on the radius arm is not the same for both the supply and exhaust valves. The former projects slightly below the latter, so that both valves can not be entirely closed at the same time. Thus when the exhaust valve is just closed the supply valve is held slightly off its seat, admitting a small quantity of oil. Similarly, when the supply valve is just closed the exhaust valve is slightly off its seat, permitting a small quantity of oil to escape.

The clearance of the radius arm between the control valves is small, about .025 inch, so that the movement of the radius arm is small, and that of the floating frame still smaller, since it is nearer the center of rotation.

On the inboard side, at each end of the floating frame, there is a steel strut which stands transversely between a socket on the floating frame and one in the end of the adjusting screw. The latter is carried in a bracket bolted to the gear box, and on the outer end there is a star adjusting wheel with a locking lug. By means of the adjusting screws the floating frame can be adjusted so as to give the correct depth of mesh as measured between the gaging rings on the gear and pinion. The purpose of the struts is to hold the floating frame against horizontal movement such as would be caused by the horizontal forces due to the interaction of the teeth. The hinging of the

struts in the sockets, permits the small compensating movement or tilting of the floating frame.

On the outboard side, at each end of the floating frame, there is a screw with a locknut, which is carried by a bracket bolted to the gear box. On the end of the screw there is a shoe which rests against the floating frame. The purpose of these screws is to prevent the teeth from getting too far into mesh.

As the floating frame is held by the inboard and outboard struts, in tilting its center line moves in a vertical plane.

Near each end of the upper bed there is a large helical steel spring held in compression between the bed and a bolt which extends downward and is screwed into the floating frame. The purpose of these springs is to balance the weight of the floating frame and pinion so there will be no pressure in the lower cylinders when the pinion is at rest. Thus both ahead and astern dynamometer gages read zero when the pinion is transmitting no power.

To prevent excessive fore-and-aft movement of the pinion due to the forces transmitted by the teeth, a dash pot is provided in the after end of the floating frame. The cylinder (brass) is secured to the floating frame, and the piston, which has two packing rings, is secured to the end of the pinion. The forward and after ends of the cylinder are connected by a passage in which there is an adjustable check valve. By adjusting the lift of the valve the dash-pot action can be regulated so as to prevent excessive end movement of the pinion.

FLEXIBLE SHAFT AND COUPLING.

The pinion is connected to the turbine shaft by a flexible shaft which passes through the axial hole and is secured at the end distant from the turbine. The flexibility of this shaft permits the tilting action of the floating frame, and also permits a slight mis-alignment of the turbine and pinion. The flexible shaft is connected with the turbine shaft by a flexible ball coupling which permits the pinion to move fore and aft freely.

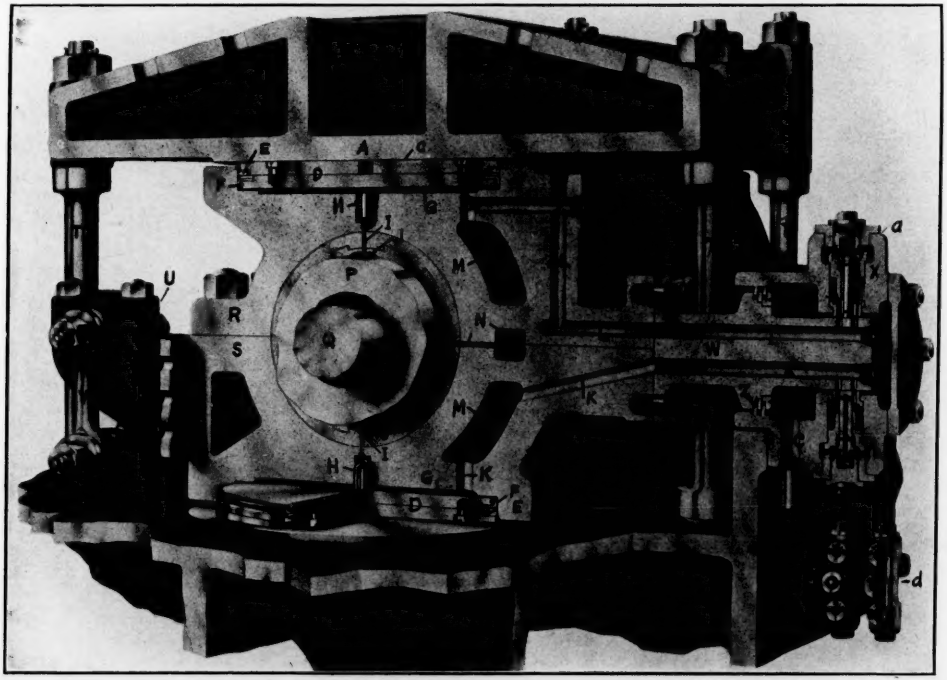


Plate III.—VERTICAL ATHWARTSHIPS SECTION THROUGH CONTROL VALVE AND FLOATING FRAME.

A, upper bed. B, lower bed. C, piston seat. D, piston. E, packing ring. F, packing leather. G, cylinder. H, check valve. I, supply channel. J, supply groove. K, ahead central part. L, astern central part. M, cored passage leading to end cylinders. N, lubricating oil-supply channel. O, pinion bearing. P, pinion. Q, flexible shaft. R, upper part of floating frame. S, lower part of floating frame. T, columns of upper bed. U, suspension bracket. V, suspension links. W, radius arm. X, central valve body. Y, cover plate. Z, packing rings. *a*, astern valve. *b*, ahead valve. *c*, drain. *d*, bridge indicating valve.

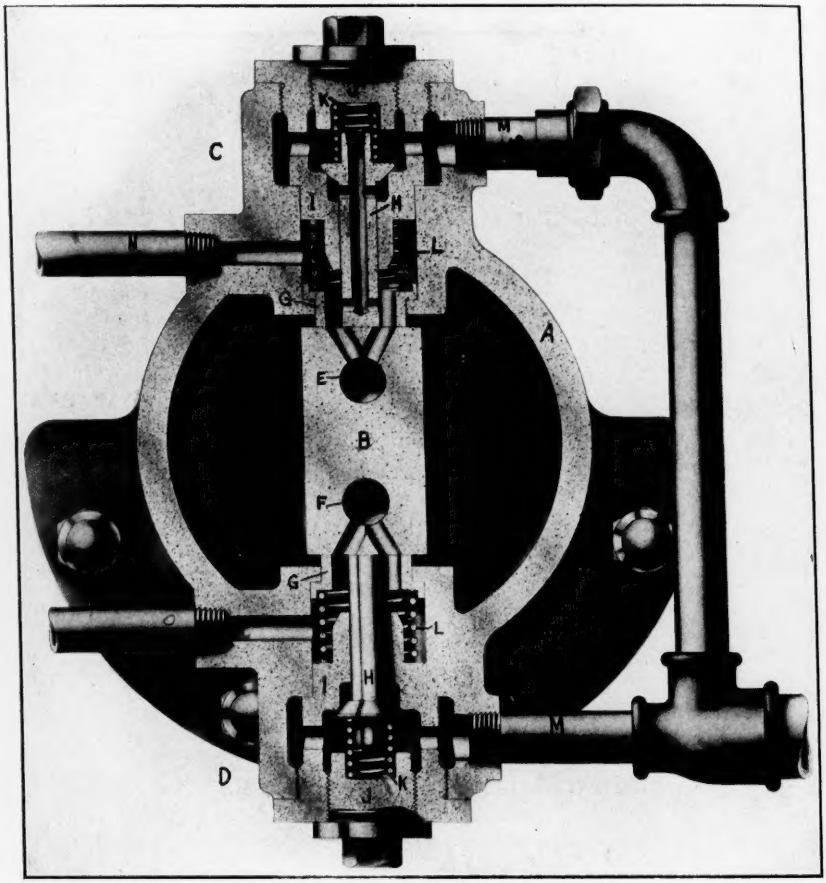


Plate IV.—Vertical Fore and Aft Section Through Control Valve of Floating Frame.

A, valve body. B, radius arm of floating frame. C, astern valve. D, ahead valve. E, astern port. F, ahead port. G, Exhaust valves. H, supply valves. I, valve seat and guide. J, cap. K, supply-valve spring. L, exhaust-valve spring. M, oil supply. N, to astern dynamometer gage. O, to ahead dynamometer gage.

The after flange of the flexible coupling, which is keyed to the pinion shaft, extends into the cylindrical recess in the forward flange which is keyed to the turbine shaft. On the outside of the after flange and on the inside of the cylindrical part of the forward flange there are corresponding semi-circular hardened-steel bushings. Together the inside and outside bushings form cylinders in which there are hardened-steel balls. On the after side of the forward flange there is a keep ring to prevent the balls from working out. As the balls roll between the inside and outside bushings, the after end of the coupling and the pinion are free to move fore and aft with respect to the forward end and the turbine shaft.

OIL SUPPLY.

From the lubricating system oil passes through the supply pipe, the vertical passage in the floating frame and enters the longitudinal supply channel. At each bearing the oil passes into the annular channels and thence through the radial holes into the longitudinal supply channels inside. From the supply channel oil passes out through the spray holes and impinges on the inboard side of the pinion. As the latter is rotating rapidly, the oil is thrown off into the pinion box and runs down into the bottom. Here it is caught up by the teeth and the current of air carried around by the pinion, and is carried into the meshing teeth. In going astern the oil is carried around over the top of the pinion. As the pinion box extends in close to the gear the oil which is thrown off of the pinion strikes the gear and is returned, so that practically all of the oil passes between the teeth. The oil from the bearings and sprays passes into the gear box and thence into the drain tank.

The oil supply for the cylinders of the floating frame is obtained from the hydraulic oil system at a pressure of about 80 pounds, and from the pinion bearings at pressure depending on the load. In the former case the oil passes through the control valve into the cylinders, and in the latter it passes directly into them through the channels leading from the bearings.

The lubricating oil which is supplied to each pinion bearing along a line of no pressure adheres to the journal and is carried around with it, forming a film between the journal and the bearing. The pressure on the film increases as it is drawn around until, near the bottom of the bearing (for ahead rotation), it reaches a line of maximum pressure, after which the pressure decreases. The bearing has an abundant supply of oil and the quantity which is drawn around by the rapidly-rotating journal is considerable. Therefore considerable oil flows into the collecting groove, which is located near the line of maximum pressure, and through the channel into the cylinder below. Thus the bearing supplies oil under pressure like a pump, or, in other words, the bearing pumps the oil.

When the pinion rotates in the ahead direction the load is on the lower bearings, which pump into the lower cylinders. As there is no pressure on the upper bearings no oil is pumped. In going astern the action is the reverse.

When stopped and starting the cylinders are supplied from the hydraulic system, as the bearing can, of course, supply none. In this case, the check valves prevent the oil from escaping.

In going ahead the pinion turns inboard, and its teeth exert an upward pressure on the teeth of the gear, resulting in a downward pressure on the pistons of the floating frame. In going astern the action is the reverse. Thus in going ahead the floating frame is supported by the lower pistons, and in going astern by the upper. In the first case the upper cylinders have no pressure, and in the second the lower ones have none.

The total pressure under the pistons is equal to the total pressure exerted by the teeth. Hence, as the area of the pistons is known, the power being transmitted by the pistons can be obtained by obtaining the unit pressure under the pistons and the r.p.m. Thus the cylinders form an accurate dynamometer for the measurement of the power transmitted by the pinion. The unit pressure in the cylinders is indicated by an accurate test gage (graduated to one pound) which is con-

nected with the valve chamber. The latter, being in communication with the cylinders, has the same pressure.

CONTROL AND ACTION OF THE FLOATING FRAME.

The control and action of the floating frame is as follows for ahead rotation. For astern rotation it is the same, except that the upper instead of the lower cylinders support the floating frame.

When, due to leakage from the cylinders, the floating frame moves downward from its mid position it hinges on the suspension link, causing the radius arm to move downward correspondingly from its mid position, so that it comes in contact with the supply-valve stem and the exhaust ring, pushing the former off its seat and seating the latter. Thus oil is prevented from escaping past the exhaust valve, and at the same time oil from the inlet chamber flows past the supply valve, through the annular space below, into the radial holes at the bottom, up through the axial hole and out through the radial holes at the top of the valve stem into the valve chamber. Thence it flows through the holes in the valve seat, through the channel in the radius arm and through the longitudinal channel in the floating frame into the cylinders. The inflow into the cylinders causes the floating frame to rise, and hinging upon the suspension link it causes the radius arm to rise correspondingly. This continues until the radius arm rises sufficiently to cause the supply valve to seat, which shuts off the supply to the cylinders and prevents further upward movement.

When, due to an excess supply of oil to the cylinders, the floating frame moves upward from its mid position it hinges on the suspension link causing the radius arm to move upward correspondingly from its mid position. This movement of the radius arm causes it to move away from the point of contact with the supply valve and exhaust ring, so that the former remains on its seat, while the latter is opened, permitting the escape of oil. Thus oil flows from the cylinders into the longitudinal channel in the floating frame, through the

channel in the radius arm, through the holes in the valve seat and into the valve chamber. Thence it escapes into the exhaust chamber and passes through the drain pipe into the gear box. The outflow of oil from the cylinders causes the floating frame to descend, and, hinging upon the suspension link, it causes the radius arm to descend. This continues until the radius arm descends sufficiently to come in contact with the exhaust ring, which shuts off the outflow of oil from the cylinders and prevents further downward movement.

Thus the control valve automatically admits oil to or exhausts it from the cylinders as required to maintain the radius arm in the mid position, and by so doing the floating frame is also maintained in the mid position.

When oil is supplied to the cylinders from the hydraulic system the exhaust valve is normally closed and the supply valve held slightly off its seat, so that there is practically a continuous supply to make up the slight leakage from the exhaust valve and the cylinders. When the oil is supplied by the bearings, the supply is considerably more than required, and consequently the quantity which escapes past the exhaust valve is considerable. This requires a considerable opening of the exhaust valve so that the supply valve normally remains closed. (The above opening is only large by comparison, as the movement of the radius arm is very small.)

The tilting action of the hydraulic floating frame is similar to that of the mechanically-supported floating frame, except that the former hinges about the center of the pinion, whereas the latter hinges about a point below.

As the pinion and cylinders are symmetrical, it follows that the total vertical force acting on the teeth of one pinion must be equal to that acting on the other; for otherwise, the greater force acting on one would produce a higher pressure in the cylinder at this end. But as the cylinders are connected through the longitudinal channel, there would be no perceptible increase, because oil would flow from this cylinder into the one at the opposite end, due to the slight difference of pressure; and, due to the displacement of the oil in the cylinder, the

floating frame would be depressed at this end until the forces acting on the floating frame and on the pinion were exactly balanced.

Since the middle of the floating frame is held in a fixed position by the control valve, when one end of the floating frame is depressed the other is elevated an equal distance. As the floating frame is held against horizontal movement by the struts, in tilting it revolves around a horizontal transverse axis passing through the center of the pinion, so that the center line of the pinion moves in a vertical plane.

In compensating for inaccuracies in the teeth or of adjustment the pinion and floating frame tilt slightly in a vertical plane as required to keep in balance the forces acting on the pinion teeth, causing oil to flow from one end cylinder into the other as required in equalizing the pressure. Thus should one part of the right-hand gear be higher than the corresponding part of the left-hand gear, the pressure between the teeth on the right-hand side will increase when this part of the gear comes into mesh with the pinion. This increase in pressure on the right-hand pinion, which is but slight, will at once cause the floating frame to tilt as required to keep the forces in balance, thus compensating for the irregularity and preventing excessive pressure on the right-hand teeth. Hence as the forces are always kept in balance by the action of the floating frame, the pressures on the teeth are always equally divided between the two pinions and equally distributed over the entire length of contact.

FEATURES OF THE WESTINGHOUSE HYDRAULIC FLOATING-FRAME REDUCTION GEAR.

The pinion is supported by liquid, and there is no direct metallic connection between the pinion and bedplate; the noise of the teeth is thereby deadened.

Only a small quantity of liquid is required to operate the pistons. Either oil or water may be used. In the latter case, provision is made to prevent the water from entering the gear

case. The maximum pressure required is about 100 pounds per square inch.

The pressure of the liquid under the pistons is indicated on an accurate spring gage, thus affording an accurate and reliable dynamometer, which indicates continuously. It requires no other instrument than a spring gage, which is familiar to and can be used by a mechanic of average intelligence. As compared with the delicate methods of doubtful accuracy and reliability used at present, the advantage of the liquid dynamometer in the measurement of power is apparent. And, further, by testing the gage, which can easily be done on board ship, this accuracy can be maintained.

Mechanically the construction of the floating frame and of the gear is simple and substantial. There are no delicate parts and all parts are easily accessible. The control valve is separate from the floating frame; it can easily be removed, and can be replaced with a spare one if necessary. The design of the valves is such that their action is positive whether they leak or not; therefore, particles of dirt under the valves will not interfere with the supply of oil to the pistons—it will simply result in a greater flow of oil. The valve chamber is separated from the gear case by packing rings, and the exhaust oil passes out through a separate drain, so that water can be used instead of oil, if necessary. For the same reason the pistons are practically sealed with the cup-leather packing, and in addition a packing ring is provided so that the piston will be reasonably tight in case the former fails.

BRIDGE INDICATING VALVE FOR DYNAMOMETER.

The purpose of this valve is to maintain an air pressure exactly the same as the oil pressure under the pistons of the floating frame. The air pressure regulated by this valve is connected to a gage on the bridge, which therefore indicates the dynamometer pressure.

Under given conditions of displacement, etc., the torque, which is a function of the dynamometer pressure, is proportional to the speed. Hence, by graduating the gage dial in

r.p.m. corresponding to dynamometer pressure, the gage will indicate the speed of the turbines. This really forms a speed indicator which shows what speed the turbines are actually making. It may be called a repeating-back signal, for after the operator on the bridge has set the turbines to make a certain speed, the dynamometer gage indicates when the turbines actually make this speed. If anything goes wrong with the bridge-control mechanism so that the turbines do not respond properly, the operator can immediately detect it by means of this indicator.

The valve consists of a bronze body which contains a cylindrical steel valve. One end of the valve is connected with the cylinders of the floating frame so that the cylinder or dynamometer pressure is always maintained in the oil-balance chamber. The opposite end is connected with the pipe leading to the bridge, and through the action of the valve air is admitted or exhausted from this end, the air-balance chamber, as required to maintain the balance—that is, to keep the air pressure the same as the oil pressure.

The cavity in the valve connects through radial holes with a groove on the outside. When this groove passes under the exhaust port in the top of the body air is exhausted from the balance chamber, and, similarly, when it passes over the supply port in the bottom air is fed into it.

Thus when the oil pressure exceeds the air pressure the greater pressure forces the valve to the left until the groove connects with the supply port. Air then flows into the balance chamber until the pressure is slightly in excess of the oil pressure when the greater pressure forces the valve to the right.

Similarly, if the air pressure exceeds the oil pressure the valve is moved to the right until the groove connects with the exhaust port, when sufficient air is exhausted to approximately balance the pressures.

Actually the valve is never in perfect balance and at rest, because there must be a slight difference of pressure to shut off the supply or exhaust after the valve has moved.

Thus when the valve is moved to shut off the supply, it continues its movement, due to the unbalanced pressure, until it reaches the exhaust position. In other words, due to the fact that the forces acting on the valve remain exactly the same after the supply is closed, it continues to move until the excess force is relieved by exhausting, thus making the full stroke. Hence the valve is never at rest, but is constantly moving back and forth through the full stroke admitting and exhausting the small quantities of air necessary to stop and reverse its movement. As the valve is a ground-fit, perfectly balanced and always well lubricated by the oil, the excess pressure required to move the valve is very small, and practically the air pressure is always equal to and balances the oil pressure.

The travel of the valve is limited by the plugs at each end.

Both the ahead and the astern valves are contained in the same casting, on which there is a bracket which is bolted to the inboard side of the gear box. The air supply is taken from the supply pipe of the bridge-control system, and is connected to a channel in the casting which leads to the supply ports of both ahead and astern valves. The exhaust from both valves also passes into a common channel, from which a pipe is led into the gear box to return any oil which may leak past the valve. The oil-balance chambers of the ahead and astern valves are connected to the ahead and astern floating-frame cylinders, respectively, through the control valves. The air-balance chambers, ahead and astern, are connected with the gage on the bridge. Quarter-inch brass piping with screwed joints is used for all connections.

The air-supply pipe is fitted with a valve, and the connections to the bridge and reduction gears are fitted with cocks. Drain cocks are fitted on the air piping on deck to drain off water which collects in pockets.

There are two valves in one body and on the bridge there is one double-hand gage for each unit (turbine and gear). The black hand indicates when the turbine is going ahead, and the red hand when going astern. Four leads of piping from the engine room to the bridge are necessary.

After calibration, the gage dials on the bridge were graduated in r.p.m., with a secondary outside scale of pressure for gage testing and correcting.

MATERIALS.

The physical and chemical characteristics of the materials used in the construction of the reduction gears are as follows:

Steel Castings.—Tensile strength, 72,000 pounds; elastic limit, 36,000 pounds; elongation, 27 per cent.; reduction of area, 45 per cent. Carbon, .275; phosphorus, .043; manganese, .600; sulphur, .028; silicon, .25.

Forging for Pinion.—Elastic limit, 51,000 pounds; tensile strength, 96,000 pounds; elongation, 24 per cent.; reduction of area, 50 per cent. Carbon, .44; manganese, .49; phosphorus, .018; sulphur, .030; silicon, .220.

Forging for Flexible Driving Shaft.—Elastic limit, 98,000 pounds; tensile strength, 120,000 pounds; elongation, 20 per cent.; reduction of area, 57 per cent. Carbon, .27; manganese, .25; chrome, 1.20; nickel, 3.21; phosphorus, .019; sulphur, .028; silicon, 1.10.

It was impossible to obtain rolled-steel rims of the required width for the reduction gears, and it was necessary to have each rim rolled in two parts. It was considered necessary to use rolled steel for this purpose. The rims are rolled by the same process as that used in the manufacture of locomotive tires.

CUTTING GEARS.

Before cutting the gear for the *Neptune* cast-iron blanks were made and the teeth were cut satisfactorily. In cutting the *Neptune's* gears it was found that cutting the oil-tempered rolled steel was very difficult, because the metal was so hard and because of the large size of the wheel. The pinions, of course, were not difficult to cut, because there are but few teeth. The hob was removed and reground before the cut was finished, and when it was replaced the remainder of the cut did not correspond.

Due to the hardness of the metal and to the lightness of the machine, there was considerable vibration while cutting. This resulted in wavy surfaces on the teeth.

It was found that the gears were not perfectly true, due to the worm wheel of the machine, which was out of true.

To remedy the above defects in subsequent gears the following was done: softer metal used in gears, hobs having more teeth were used, the hob was run all the way through, making a complete cut before regrinding. The worm wheel was made perfectly true. Also, a heavier machine has been obtained for the larger work.

The gears which have been cut lately are apparently perfectly true, which shows in the remarkably smooth running.

Due to the inaccuracies of cutting the gears, they do not run as smoothly as they might, and probably some of the pitting which occurred at the shop was due to this. The irregularities of the gear are very perceptible while running, particularly when running slowly.

DASH POT.

While testing the gears at East Pittsburgh it was found that there was considerable end movement of the pinion. This was due principally to the fact that the gear was not held against end movement by a suitable thrust bearing. The end movement, which was small in starting, was augmented by the swing or inertia of the parts until it was considerable. To overcome any difficulty which might be caused by this, dash pots with adjustable valves were fitted on the ends of the pinions. The dash pots reduced the end movement considerably.

It is doubtful whether the dash pots are required in service, because the gears are held against end movement by the thrust bearings and the horizontal forces between the teeth prevent excessive movement of the pinion.

While running there is only a slight end movement of the pinion (not over 1-16 inch) when the check valve of the dash pot is prevented from working.

FLEXIBLE COUPLING.

The ball couplings are lubricated with vaselene and graphite. The ball sockets are filled with this lubricant, as a matter of routine, when the machinery is secured after steaming. The flexible ball coupling of the reduction gear has run very well up to the present time. It is perfectly balanced and there is perfect freedom of end movement of the pinion which moves back and forth continually while running. There is no perceptible wear in the balls and bushings. As a whole this coupling appears to be a particularly good type. It is far better than the original link coupling, which could not be balanced properly.

WEAR AND BACKLASH.

There appears to be no appreciable wear in the teeth and no backlash can be detected in running. When reversing there is no jar or noise. Without observing the movements of the machinery it is impossible to tell when the gear reverses.

FINAL REPORT ON REDUCTION GEARS.

The reduction gears have run particularly well under all conditions and have given no trouble whatever. While running they require practically no attention. All repairs, overhauling, adjusting, etc., can easily be done by the ship's mechanics. The gears make more noise than is usual in an engine room, but it is neither disagreeable nor excessive. The gears make about the same noise, or less than a high-speed turbo-generator of the same power. The writer does not consider noise as a serious objection to the use of gears for naval vessels. There is no perceptible vibration due to the gears.

The operation of the gears in heavy weather is entirely satisfactory. During one voyage the propellers came out of water frequently, and at times they were almost entirely out. The governing mechanism prevented excessive racing. Under these conditions the gears performed satisfactorily and appeared to be under no unusual strain. The gears have been

subjected to shocks due to heavy weather and due to the port propeller striking an obstruction without damage to the teeth and flexible shaft. Greater shocks may be experienced, but, in my opinion, the gears will prove amply strong to withstand any shock which can be transmitted by the propeller shaft as both the teeth and the flexible shaft are considerably stronger, besides being protected by the fly-wheel action of the gear. In this respect I consider the gears entirely satisfactory for naval use.

In maneuvering, the gears have performed satisfactorily in every way. There is no perceptible backlash or jar when reversing.

The gear box was not oiltight, due to changes that were made. In future work the gear box should be made perfectly tight. The oil casing between the reduction gear and the turbine is not tight, and considerable oil has been lost from it. It is also inaccessible. The shaft, governor-driving gear and flexible coupling should be completely enclosed in an oiltight casing. It is recommended that this casing be made of cast iron, divided horizontally, and with the cover free for removal.

The accessibility and overhauling facilities of the gears are excellent in most respects. No gear is provided for lifting the gears. This is essential on a naval vessel. Separate lifting gears should be provided for the gear and for the floating frame. Part of the working platform is located above the gear and has to be removed before the cover can be lifted. This defect should be avoided in future work. Suitable socket wrenches and lifting eyes for all parts should be provided. The plug gages for adjusting the mesh of the gears are a great convenience, if not a necessity, and should always be provided. As the gears can be seriously damaged by not having the struts adjusted correctly, it is recommended that caps be provided to completely enclose the adjusting heads and locking devices. The cover has to be lifted to remove the main-bearing caps. This should be avoided if possible. It is considered desirable to fit liners between the brasses so that the

bearing clearance can be adjusted without planing off the brasses.

The operation of the hydraulic floating frame has been entirely satisfactory. The pressure has been distributed uniformly over the entire length of the teeth, and there has been no indication of intense local pressure. The automatic-control valve has been thoroughly reliable and has given no trouble. The flexible ball coupling has worked satisfactorily and is superior to the coupling originally designed. The original packing leathers are still in use and will probably last for a long time, as there is little wear on them. The oil supply from the bearings is reliable and entirely satisfactory. The adjustments to the floating frame have been easily made by the ship's mechanics.

In the writer's opinion, the floating frame or similar device for aligning the pinion automatically is absolutely essential for large gears. Ship's mechanics could not be relied upon to make the extremely accurate alignment that would otherwise be required. For military reasons it is of great importance to be able quickly to overhaul and assemble with the absolute assurance that the machine will run perfectly and be absolutely dependable. This feature of the floating frame is of utmost importance in naval work, where self-maintenance and reliability are essential. With the floating frame it is only necessary to adjust the struts according to the plug gages, which any mechanic of ordinary ability can do. The alignment takes care of itself.

There has been practically no wear. The pinion teeth are highly polished and show no signs of cutting. The gear teeth are just barely polished along the pitch line. At the present rate of wear the life of the gears will be considerably longer than that of the ship. The pinions will undoubtedly last five or six years at the least under service conditions. It will be comparatively easy and inexpensive to replace the pinions if it becomes necessary. Although it will probably never be required, the gear rims containing the teeth can be replaced with but little difficulty. The bearings show very little wear. The

The writer considers the reduction gear entirely satisfactory for use on naval vessels and recommends its use where it is found to be advantageous.

PERFORMANCE OF BRIDGE-CONTROL SYSTEM.

(See Plates V, VI and VII.)

During the trial of the *Neptune,* and until the bridge-control system was modified to eliminate the defects which existed when it was first installed, it did not work satisfactorily. Most of the troubles which were experienced were caused by faulty installation and were not due to defective action of the mechanism. Since correcting the defects the bridge-control system has worked well and given excellent results.

The turbines can be handled in the engine room or from the bridge, both operating stations being essentially the same. It makes no difference which station is used, as the turbines respond exactly the same to the bridge operator as they do to the engineer. Both use the same system, but control it from different places. There could be three or even more operating stations if desirable. In a battleship, for example, a third station could be located in the central station of the ship.

Operating the turbines in the engine room is much easier than with the usual types of turbine and engine-operating gear. There are two small levers, one for each turbine, which are operated by one man without difficulty. These two levers automatically control all of the necessary functions with a rapidity and precision which is impossible with manual control. The three principal functions controlled by the operating levers are reversing, regulation of speed, and nozzle control. The regulation of speed refers to the actual r.p.m. the operator desires, and not the fixed opening of a valve, which only serves as the means for the operator to do this.

Due to the action of the mechanism, reversals and changes of speed are very rapid, for when the operating lever is moved from one position to another, the mechanism automatically opens wide or closes completely the turbine-inlet valve, which

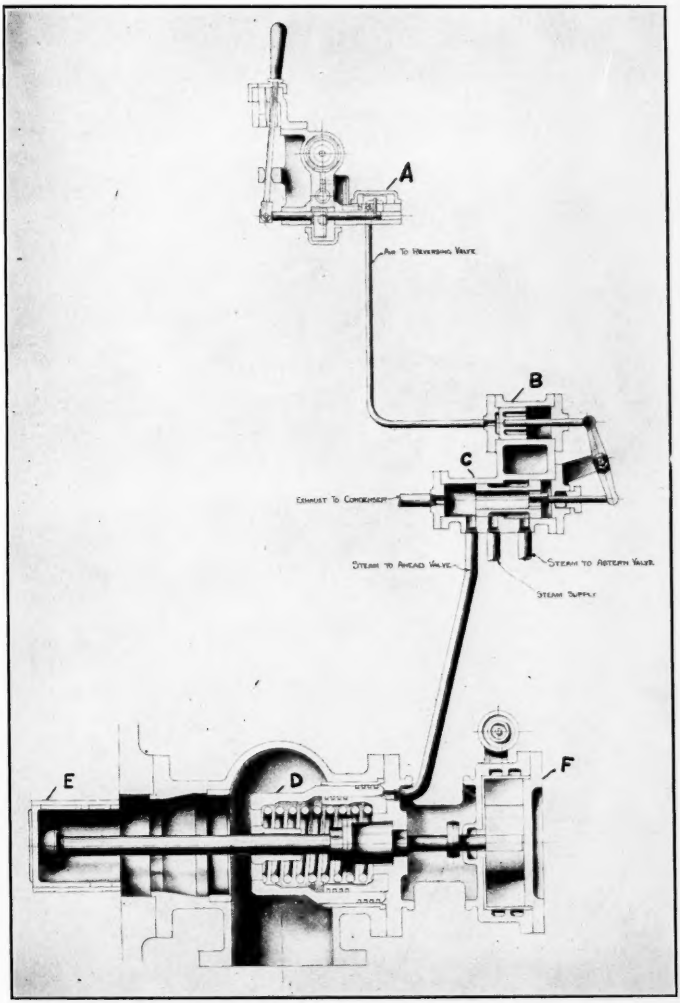


Plate V.—DIAGRAM OF BRIDGE-CONTROL MECHANISM, SHOWING THE PARTS USED FOR REVERSING.

A is the air-reverse valve of the control valve. B is the air-operating cylinder of the steam-reverse valve. C is the steam-reverse valve. D is the turbine-operating or poppet valve. E is the nozzle control valve; F, its hydraulic-operating cylinder. E and F are not used in causing reversal.

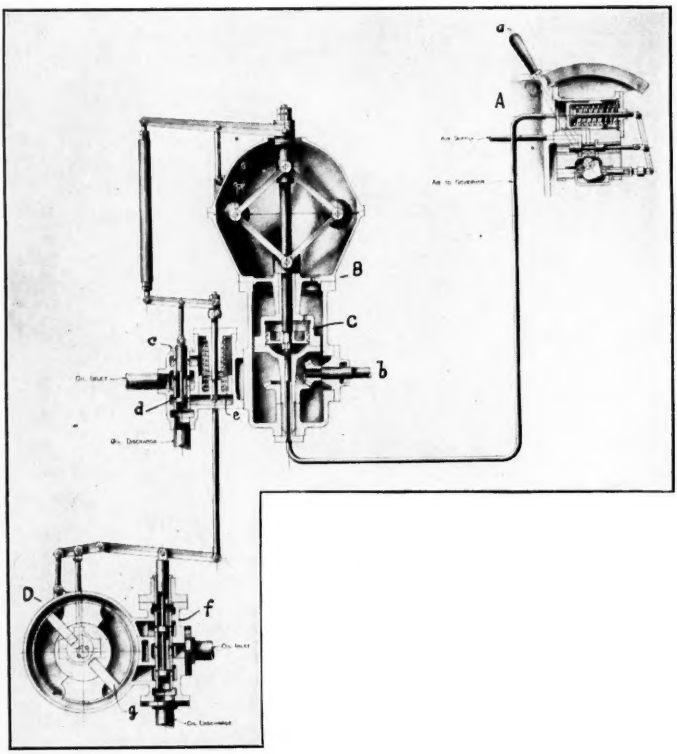


Plate VI.—Diagram of the Bridge Control Mechanism Showing the Parts Used for the Regulation of Speed.

A is the regulating part of the air-control valve. B is the turbine governor. C the governor relay. D the hydraulic-operating cylinder. E (shown in Plate V) is the nozzle control valve. *a* is the operating lever with which the speed is controlled. *b* is the governor driving shaft which is geared to the turbine. *c* is the air cylinder and balance piston of the governor. *d* is the pilot valve of the governor relay and *e* is its cylinder and piston. *f* is the double-pilot valve of the hydraulic-operating cylinder, and *g* the double-acting wing piston.

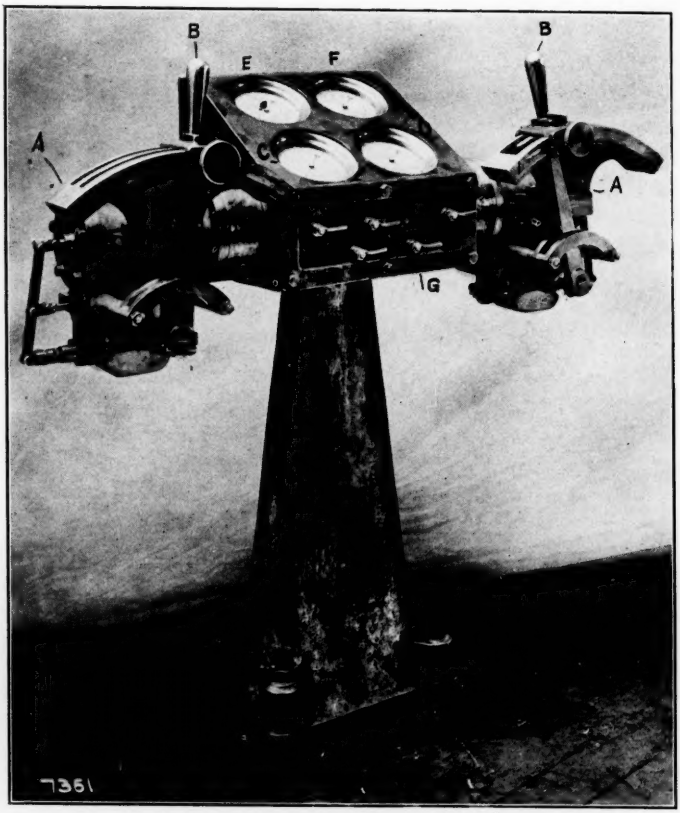


Plate VII.—THE BRIDGE STAND SHOWING THE AIR-CONTROL VALVES A FOR BOTH TURBINES.

Regulation of speed and reversal are controlled by the operating levers B. C is the main steam-pressure gage. D the air-reservoir pressure. E are indicators showing the governer air pressure and the revolutions of the propellers. G are cut out cocks for shutting off the control valves as for connecting both turbines to one valve so as to obtain exactly the same speed on both turbines. This stand is located on the bridge and is used in the same manner as the usual mechanical telegraphs. The control valves are connected with the engine room by ¼-inch pipes as shown in Plate VI. The engine-room operating stand is practically the same as the one on the bridge, the pipes being connected in parallel through a single-lever operated cut-out valve.

remains so until the exact speed for which the lever is set is attained. This feature of rapid response of the turbines is of importance, because it causes the ship to respond in the minimum of time. This action is always the same. It gives the ship uniform handling qualities, and, what is more important, it can be depended on in an emergency.

One of the officers who saw the operation of the system suggested that when steaming under reduced boiler power this feature might cause the steam pressure to fall too low, particularly when operating from the bridge. The gage on the bridge keeps the operator informed of the steam pressure, and as the "low" pressure is plainly marked on the dial, there would be no excuse for allowing the pressure to fall below this point except in case of emergency, in which case it would be done intentionally. With reduced boiler power, main engines of any kind must be operated at reduced speed. The bridge operator has all of the information necessary for operating the turbines, and if the pressure falls too rapidly, all that is necessary is to reduce the speed. This condition has been simulated here a number of times when the boilers were generating steam slowly, and there has been no difficulty in either bridge or engine-room operation.

When the operating lever is set for a certain speed the turbine comes up to this exact speed and it is maintained automatically. This is true under all conditions regardless of steam pressure, vacuum, propeller resistance, etc., provided the speed is not too great—*i.e.*, below the maximum possible under the conditions. This feature makes the operation easy and precise. There is no guesswork as to speed and no revolutions to be counted while maneuvering. This is a valuable feature for either bridge or engine-room operation. It makes it possible for one man to do what several are required to do with the usual operating gears, and it is done better.

While steaming, the operating lever is set for the required speed, which is maintained as noted above. It is not necessary for the engineer to stand by the throttle and constantly to regulate the speed. In effect, the human governor is replaced

by a mechanical one which is more accurate and better. The engineer of the watch is free to go about the engine room and see that the machinery is running properly. All that is necessary is that he or some one on watch be near enough to the operating station to answer a signal promptly. This is true when operating from either station. Compare this mechanical regulation with the method employed on naval vessels steaming in formation. Two men, at least, constantly timing and adjusting the revolutions, and nothing else. These two men could be removed entirely or used to better advantage superintending the machinery, which is usually left to less experienced men. The advantage of the mechanical regulation is obvious. This feature of the control system is important, and even if the engines were operated from the engine room only would render it desirable.

As noted, after correcting the defects the operation of the control system has been satisfactory. There is practically no difference in operating on the bridge or in the engine room. The turbines respond and reverse quickly, and the ship responds quickly. Handling the turbines is easy and accurate.

The bridge station has been used in steaming, in going alongside ships and in docking. While steaming it requires practically no attention, as the regulation is automatic. The speed has been regulated accurately, but not as accurately as desired. Unfortunately, there has been no opportunity for steaming in formation, but I have no doubt of the suitability of the control system for this purpose. In maneuvering, in coming to anchor, going alongside, etc., the regular bridge detail has handled the turbines without difficulty. In this connection attention is invited to the fact that none of the deck officers is experienced in engineering.* In fact, they have no knowledge of it. If men of this type can use the bridge control, there should be no difficulty on a naval vessel. In maneuvering, the control gear has operated satisfactorily. The indicators show the operator what the turbines are set to do and what they are doing. Gages show the air and steam

* The *Neptune* is manned by a merchant crew.

pressure available. The gages and indicators have special dials so that the indications can be understood by an untrained man. With this control gear it is as easy for one man to operate the engines of a battleship as it is to run an automobile.

Under service conditions there is a saving in time in operating from the bridge of from 2 to 6 seconds, which depends principally on the engine-room operator. Under test conditions of course this would be less. To the officer on the bridge it gives confidence to know that the propellers are entirely under his control, that the control gear operates quickly and exactly the same each time. There is no doubt as to whether the signal is being answered properly and promptly. There is no one over whom he has no control to use discretion to keep the steam up when the ship is in a dangerous position. In case of emergency the officer on the bridge can best decide whether it is best to save steam or a collision.

In two instances it was necessary to make reserve speed to avoid collision. The steam pressure dropped very low and some of the auxiliaries were about to stop. The engineer protested vigorously against such unwise operation. He, not knowing the danger, would probably have eased up to save steam, and collision would have resulted.

During one voyage the propellers came out of water so that excessive racing would have occurred with ordinary machinery. Most of the time they came half way out and frequently they came practically all the way out. Due to the action of the governing mechanism racing was prevented. The speed increased and decreased slightly—about four per cent. estimated. The turbine-inlet valves were opening and closing continuously and at times they were entirely closed. Quick-operating valves are necessary effectively to prevent racing. This feature of the control system is of importance because it will enable a vessel to steam at considerable speed in heavy weather.

After being modified the control system has been thoroughly reliable. Even if the bridge control should fail it only requires a few seconds to change to the engine room. This feature has been frequently tested and found satisfactory. However,

with the improvements which will be made in accordance with the experience gained with this the first installation, the writer is convinced that operation from the bridge will be absolutely positive and reliable under all conditions. It will probably be more reliable than the telegraph, although the latter is good in this respect. As the general principles are the same, there is no reason why it should not be as reliable as the air brake, or even more so, as the service is less severe.

The governing part of the mechanism has been reliable and has given no trouble from the beginning. As this part is essentially the same as governors in use on shore stations it is considered thoroughly reliable and satisfactory for use on board ship.

Provided the control gear is operated properly, which is not difficult, there is no danger in bridge operation. The instructions require the turbines to be ready for starting before the bridge station is connected. If an accident is likely to occur in the engine room the engineer can cut out the bridge and slow down. As far as accidents to the machinery are concerned, it makes no difference which operating station is being used; the turbines run the same, and if necessary the engineer can slow down or stop in either case.

Both turbines can be controlled from one valve to obtain exactly the same air pressure under the pistons of both governors. This was used while steaming when it was desirable to keep the speed of both turbines the same. In steaming in formation this would be useful, because the speed of both turbines can be adjusted more easily with one valve.

It is believed that the bridge-control system will prove very useful in steaming in formation, because the speed is automatically regulated with accuracy, and it is most convenient to make frequent changes or adjustments of speed. It should be very easy to keep the ship in exact position. Complete control of the machinery, as well as of the steering, would be an advantage for both precision and safety.

In battle the engines could be controlled from the conning

tower or from the central station. In case any part of the control system is shot away, the air pressure in the reservoir and the speed of the turbines will decrease. The engineer would at once know the cause and change to the engine-room station. The danger of this would be remote if operating from the central station.

There has been no opportunity of steering the ship with the propellers, but it is believed that this could be done without difficulty. One turbine would be set for a constant speed; the speed of the other would be increased or decreased as required. The lever of the valve used for steering would be used in exactly the same manner as a helm; that is, if the starboard valve was being used it would be moved to starboard to cause the bow to move to port, and vice versa. This feature would be an advantage at any time as an emergency-steering system.

A properly-designed bridge-control gear will, in the writer's opinion, be entirely satisfactory and reliable for naval vessels, and it will be desirable because of the features set forth in the foregoing.

This system could be used with reciprocating engines as well as with turbines.

PERFORMANCE OF TURBINE-CONTROL GEAR DURING OFFICIAL TRIAL.

(From Report of Trial Board, July 29, 1911.)

The ship was going ahead, the engines were stopped and the bridge station was connected. The engines were started ahead and then reversed and started ahead three times. The data taken is given in the appended table.

After making the foregoing tests the engine-room telegraphs were put full-speed astern. This was a prearranged signal to indicate that the bridge control gear was not in operation and that the engines were to be worked from the engine room. This arrangement was unknown to any one in the engine room. It required 15 seconds to shift from the bridge to the engine-room station.

During the above tests the engines responded quickly to the bridge operation.

TEST OF BRIDGE-CONTROL GEAR DURING OFFICIAL TRIAL.

(From Report of Trial Board, July 29, 1911.)

Test number.	1	2	3	4	5◁
Turbines operated from	E.R.	Bridge.	B	B	E.R.
Speed ahead before reversing	13	...	...	...	...
*Began turning astern, starboard	0:17.4	0:14	0:13.2	0:16.6	0:22
*Began turning astern, port	0:32.5	0:14.4	0:15	0:16.6	0:22
*Full speed astern, starboard	2:14	1:32	1:07	1:06.5	1:10
*Full speed astern, port	2:35	1:40	1:15	1:10	1:10
R.P.M., astern	...	85	75	75	70
†Began turning ahead, starboard	0:11.2	0:14	0:15	0:14	...
†Began turning ahead, port	0:13.4	0:16.4	0:15.4	0:16	...
†Full speed ahead, starboard	1:33	1:32	1:45	1:11	...
†Full speed ahead, port	1:33	1:40	1:45	1:11	...
R.P.M. ahead	...	105	100	90	...
	□				

* From time telegraph as operating lever was thrown full-speed astern.
† From time telegraph as operating lever was thrown full-speed ahead.
□ From signal to dead in water, 4 minutes, 33 seconds.
◁ Time to change stations from bridge to engine room, 15 seconds.
◁ Times in test No. 5 taken from signal to change stations.
Times in minutes and seconds

NEW RULES FOR THE INTEGRATION OF CURVES.

By MARK FARGUSSON.

Much has been written in this country in support of the Trapezoidal Rule for the integration of ship curves, it being maintained that its simplicity, flexibility and ease of manipulation more than counterbalance its admitted inaccuracy. On the other hand, while it is agreed that Simpson's Rule gives results much more nearly correct, it is held that this additional accuracy involves an amount of labor quite out of proportion to its value.

It is the purpose of this paper to propose two rules which overcome the objections to each of the above; which combine the accuracy of the latter with the simplicity of the former. In order to distinguish them I will call them respectively the Proposed First Rule and the Proposed Second Rule.

THE PROPOSED FIRST RULE.

If we let

y = the lengths of the ordinates at the sections denoted by their subscripts;

n = the number of equal parts into which the axis of abscissae is divided along the length of the curve;

i = the length of the common interval between the ordinates; and

A = the computed area of the curve: then this rule may be expressed as follows:

$$A = i\left\{\frac{y_0}{2} + y_1 + y_2 + \ldots + y_{n-1} + \frac{y_{n-1}}{2}\right\}$$

In other words, it differs from the Trapezoidal Rule in that the first and last ordinates are taken not at the ends of the curve but at a distance of $\frac{i}{4}$ therefrom.

The data are no more difficult to take off than for the Trapezoidal Rule and the calculation is as simple, while, besides its

very greatly increased accuracy, it has the further advantage of giving a more rational solution for the determination of moments with a corresponding improvement in results.

The principles involved in this rule and the reason for its greater degree of accuracy may be easily seen by glancing at Figure 1, which shows a parabolic curve representative of the section of a ship. Here by the Trapezoidal Rule the area of the curve would be given as the sum of the areas of the following rectangles:

$$d\,j\,k\,g = i y_1,$$
$$h\,m\,n\,k = i y_2,$$
$$l\,g\,r\,n = i y_3, \text{ and}$$
$$o\,s\,u\,r = \frac{i y_4}{2}.$$

That portion of the curve shown as $a\,c\,e\,g$, which in the Proposed Rule is represented by $a\,b\,f\,g = i\frac{y_{\frac{1}{2}}}{2}$, is in the Trapezoidal Rule altogether omitted, since $y_0 = 0$, and hence $\frac{i y_0}{2} = 0$. Further than this $\frac{i y_4}{2}$ is too great, since it represents the area $o\,s\,u\,r$ instead of $p\,t\,u\,r$, as it should and does in the Proposed Rule in the term $\frac{i y_{n-\frac{1}{2}}}{2}$. It may be said that the excess in the area $o\,s\,u\,r$ tends to neutralize the omission of the area $a\,c\,e\,g$, and while this may, to a certain extent, be so, it is wrong in principle and the area is not properly proportioned among the ordinates, so that when, in the calculation of moments, the partial areas are multiplied by their respective arms the discrepancy may easily be so magnified as to involve serious errors, since the factor of the former is $n\,i$ while that of the latter is 0.

But besides the greater accuracy in the total area of the curve and the more correct distribution of that area about the several ordinates, the Proposed Rule has the still further source of accuracy in the calculation of moments by its more correct location of the ordinates for obtaining the proper length of arms.

This is shown clearly in Figure 1, where it is seen that in rules using ordinates y_0 and y_n the moment of $a\,c\,e\,g$ is entirely omitted, and would be so omitted even if y_0 had a positive value, since the arm of y_0 is 0. Further, for the moment of the area $o\,s\,u\,r$, $n\,i$ would be taken for the arm which is to the far edge of the area and not to the center of gravity. On the other hand, in the Proposed Rule, not only is the area of $a\,c\,e\,g$ closely approximated to in the area $a\,b\,f\,g = \frac{i\,y_{\frac{1}{2}}}{2}$, but the arm has a positive value of $\frac{i}{4}$, thus giving for the moment, $\frac{i^2 y_{\frac{1}{2}}}{8}$. Then, too, for the area $v\,s\,u\,r$, closely approximated to by the area $p\,t\,u\,r$ represented by $\frac{i\,y_{n-\frac{1}{2}}}{2}$, the arm would be $(n - \frac{1}{4})i$, which is very close to its true value. To show to what extent these items affect the accuracy of the results it may be noted that the errors in the determination of the vertical center of gravity of the Curve A shown in Figure 2 is, by the

Trapezoidal Rule, 1.23 per cent., while by the
Proposed First Rule, 0.29 per cent.

PROPOSED SECOND RULE.

As there may be certain calculations where it simplifies the work by having the ordinates taken in the usual way at the ends of the curves, I would offer a Second Rule for use in such cases. This may be expressed as follows:

$$A = i\left\{\frac{y_0 + \frac{y_1 - y_0}{c}}{2} + y_1 + y_2 - \ldots + y_{n-1} + \frac{y_n + \frac{y_{n+1} - y_n}{c'}}{2}\right\}.$$

While this looks most formidable it is not at all so in fact, as its use will soon show. If in this formula we let $c = c' = 4$, which represents the average conditions where curves of all classes are to be met, it should compare favorably with Simpson's Rule and give results sufficiently accurate for all practical purposes. This value of 4 is that towards which these

terms approach as the curves treated become flatter, either through their nature or through a division into smaller intervals. It is that with which the curves shown in Figure 2 are calculated in the Tables. However, in curves such as these, which have a considerable degree of curvature, the accuracy of this rule will be very greatly increased, until, in fact, it is practically that of the First Rule, if the values of c and c' are chosen for the particular curve, or class of curves, being treated. While the range of values through which these terms may pass is rather great the selection of the right one involves no particular labor, as it may be done by simple inspection. In many cases it will be found that it would be of advantage thus to determine only one of these values and that the other, owing to the flatness of the curve at that point, may very well be taken as 4, its mean value. It may also be, where the curve is the same at both ends as in an ellipse, that one value will do for both. To illustrate the effect of a more careful selection of these values, let us take Curve A, for example. As calculated in the Table with $c = c' = 4$, the average values, the error is 0.430 per cent., while if c is taken as 2, leaving $c' = 4$, it will be reduced to 0.165 per cent. The selection of these special values of c and c' will, however, rarely be necessary, as by taking the average value of 4 for each of them results may be had which in general will be of quite sufficient accuracy and which will compare favorably with those obtained by Simpson's Rule.

In the calculation of moments, even when the Second Rule is used, it is well to take the first and last arms as $\frac{i}{4}$ and i $(n - \frac{1}{4})$ respectively.

In cases of such calculations as those of integrating to successive water lines, it will be found of advantage to combine the two Rules, using for the initial ordinate $y_{\frac{1}{2}}$ and taking the other at the even water lines, thus:

$$A = i\left\{\frac{y_{\frac{1}{2}}}{2} + y_1 + y_2 + \ldots + y_{n-1} + \frac{y_n + \frac{y_{n-1} - y_n}{c'}}{2}\right\}.$$

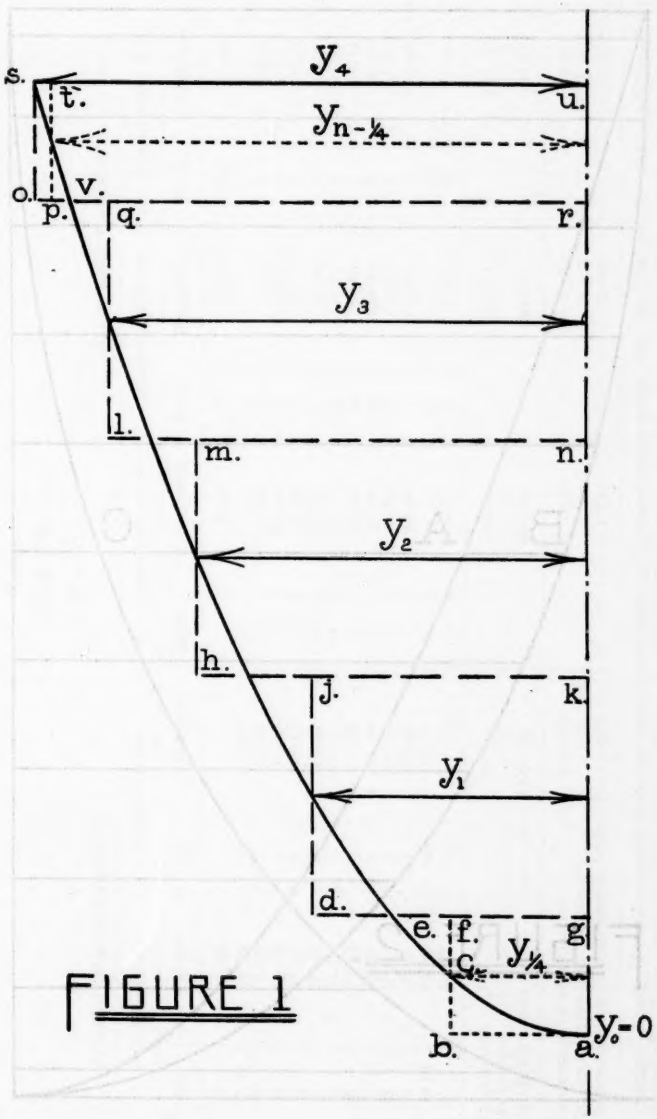


FIGURE 1

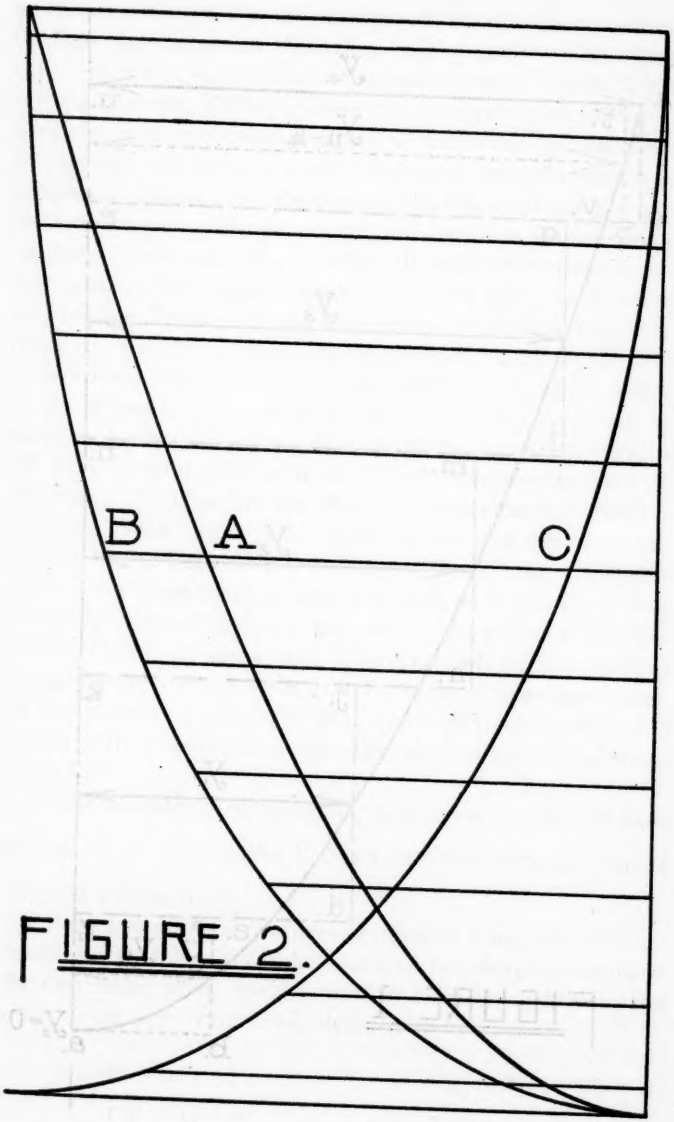


FIGURE 2.

TABLE I.

CURVE A.

$y = 1\sqrt{18x}$.

Interval between ordinates = 5. Actual mathematical area = 1,000.

Ordinates.		First Rule.		Second Rule.		Simpson's Rule.		Trapezoidal Rule.	
No.	Value.	Mult.	Functions.	Mult.	Functions.	Mult.	Functions.	Mult.	Functions.
0	0.00	...	.....	...	1.19	1	0.00	$0\frac{1}{2}$	0.00
0.25	4.74	$0\frac{1}{2}$	2.37	...	.....	...	.....	...	.....
1	9.49	1	9.49	1	9.49	4	37.96	1	9.49
2	13.42	1	13.42	1	13.42	2	26.84	1	13.42
3	16.43	1	16.43	1	16.43	4	65.72	1	16.43
4	18.97	1	18.97	1	18.97	2	37.94	1	18.97
5	21.21	1	21.21	1	21.21	4	84.84	1	21.21
6	23.24	1	23.24	1	23.24	2	46.48	1	23.24
7	25.10	1	25.10	1	25.10	4	100.40	1	25.10
8	26.83	1	26.83	1	26.83	2	53.66	1	26.83
9	28.46	1	28.46	1	28.46	4	113.84	1	28.46
9.75	29.62	$0\frac{1}{2}$	14.81	1	.....	...	.....	...	.....
10	30.00	...	.....	...	14.80	1	30.00	$0\frac{1}{2}$	15.00
Summation			200.33		199.14		597.68		198.15
Multiplier			5		5		$\frac{5}{3}$		5
Calculated area			1,001.65		995.70		996.13		990.75
Percentage of actual error			+0.165		−0.430		−0.387		−0.925
Errors relative to First Rule			1.000		2.606		2.345		5.667
Errors relative to Simpson's Rule...			0.426		1.111		1.000		2.390

NOTE.—c and c' in the Second Rule are each here taken at 4, the average value, but not the best for the conditions here illustrated. Here c should have been 2. c' may as well remain 4.

TABLE II.

CURVE B.

$$y = 0.8\sqrt{100x - x^2}.$$

Interval between ordinates = 5. Actual mathematical area = 1,570.80.

Ordinates.		First Rule.		Second Rule.		Simpson's Rule.		Trapezoidal Rule.	
No.	Value.	Mult.	Functions.	Mult.	Functions.	Mult.	Functions.	Mult.	Functions.
0	0.00	...	.....	...	2.18	1	0.00	$0\frac{1}{2}$	0.00
0.25	8.89	$0\frac{1}{2}$	4.44	...	.....	...	.....	...	.....
1	17.44	1	17.44	1	17.44	4	69.76	1	17.44
2	24.00	1	24.00	1	24.00	2	48.00	1	24.00
3	28.57	1	28.57	1	28.57	4	114.28	1	28.57
4	32.00	1	32.00	1	32.00	2	64.00	1	32.00
5	34.64	1	34.64	1	34.64	4	138.56	1	34.64
6	36.66	1	36.66	1	36.66	2	73.32	1	36.66
7	38.16	1	38.16	1	38.16	4	152.64	1	38.16
8	39.19	1	39.19	1	39.19	2	78.38	1	39.19
9	39.80	1	39.80	1	39.80	4	159.20	1	39.80
9.75	39.99	$0\frac{1}{2}$	19.99	...	.....	...	.....	...	.....
10	40.00	...	.....	...	19.98	1	40.00	$0\frac{1}{2}$	20.00
Summation.....			314.89		312.62		938.14		310.46
Multipliers.....			5		5		$\frac{5}{3}$		5
Calculated area.....			1,574.45		1,563.10		1,563.57		1,552.30
Percentage of actual error.....			+0.232		−0.490		−0.460		−1.178
Errors relative to First Rule.....			1.000		2.112		1.983		5.078
Errors relative to Simpson's Rule ...			0.504		1.065		1.000		2.561

NOTE.—c and c' in Second Rule are each here taken at 4, the average value, but not the best for conditions here illustrated, in which c should equal 2. c' may well be left as 4.

TABLE III.

CURVE C.

$$y = 40 - 0.8\sqrt{100x - x^2}.$$

Intervals between ordinates = 5. Actual mathematical area = 429.20.

Ordinates.		First Rule.		Second Rule.		Simpson's Rule.		Trapezoidal Rule.	
No.	Value.	Mult.	Functions.	Mult.	Functions.	Mult.	Functious.	Mult.	Functions.
0	40.00	...	.....	...	17.82	1	40.00	$0\frac{1}{2}$	20.00
0.25	31.11	$0\frac{1}{2}$	15.56	...	.....	...	.....	...	.....
1	22.56	1	22.56	1	22.56	4	90.24	1	22.56
2	16.00	1	16.00	1	16.00	2	32.00	1	16.00
3	11.43	1	11.43	1	11.43	4	45.72	1	11.43
4	8.00	1	8.00	1	8.00	2	16.00	1	8.00
5	5.36	1	5.36	1	5.36	4	21.44	1	5.36
6	3.34	1	3.34	1	3.34	2	6.68	1	3.34
7	1.84	1	1.84	1	1.84	4	7.36	1	1.84
8	0.81	1	0.81	1	0.81	2	1.62	1	0.81
9	0.20	1	0.20	1	0.20	4	0.80	1	0.20
9.75	0.01	$0\frac{1}{2}$	0.00	...	.....	...	.....	...	.....
10	0.00	...	.....	...	0.03	1	0.00	$0\frac{1}{2}$	0.00
Summation			85.10		87.39		261.86		89.54
Multiplier			5		5		$\frac{5}{3}$		5
Calculated area			425.50		436.95		436.43		447.70
Percentage of actual error.....			−0.862		+1.806		+1.685		+4.310
Errors relative to First Rule..			1.000		2.095		1.955		5.000
Errors relative to Simpson's Rule.....			0.512		1.072		1.000		2.558

NOTE.—As with the other curves the value of $c = 4$ used in this table is not the best for the conditions here illustrated, where 2 would be nearer correct. c' may as well remain equal to 4.

In order to illustrate the simplicity of manipulation of these Rules as well as their accuracy as compared with Simpson's and the Trapezoidal Rules, I have calculated by each the area of each of the following curves, shown in Figure 2:

Curve A, where $y = \sqrt{18x}$,
Curve B, where $y = 0.8\sqrt{100x - x^2}$, and
Curve C, where $y = 40 - 0.8\sqrt{100x - x^2}$

The results of these calculations are shown in the Tables and need no explanation. It is only necessary to bear in mind that the value of c here used in the Second Rule is not that best adapted to the conditions here obtaining, where c should in each case have had a value of 2. Had this been done, as would have been the case in actual practice, results comparable to those of the First Rule would have been obtained.

Of course, greater accuracy could be obtained by any of these rules, although at the expense of considerable labor, if the end divisions were subdivided or if the number of ordinates were increased, say from 11, as used in these calculations, to 21, as is usual in most practical work; but my idea has been to show that even with a small number of ordinates the Proposed Rules will give results which approximate very closely to the true values. On the whole, therefore, it would seem that their combined accuracy and simplicity should make them by far the most desirable rules for the purpose of ship calculations.

As this paper has been most hurriedly prepared, the writer trusts that he will be pardoned if any errors have escaped his attention.

NOTES.

A NEW ANALYSIS OF INDICATOR CARDS.

ADAPTED FROM A PAPER BY THE ORIGINATOR OF THE ANALYSIS, J. PAUL CLAYTON.

DESCRIBES MEANS FOR DETERMINING FROM AN INDICATOR DIAGRAM AN ALMOST COMPLETE ANALYSIS OF THE CYLINDER PERFORMANCE OF A RECIPROCATING ENGINE, INCLUDING RESULTS HERETOFORE CONSIDERED IMPOSSIBLE TO SO OBTAIN ACCURATELY.

If when fluid expands, as does steam in the cylinder of an engine, its pressure varies inversely as its volume, the relation of pressures and volumes will be represented by the middle curve in Fig. 1. The product of the volumes and the pressures corresponding with the several volumes will be as follows:

Vol.	×	pressure	=	constant.
1	×	120	=	120
2	×	60	=	120
3	×	40	=	120
4	×	30	=	120
5	×	24	=	120
6	×	20	=	120

Setting these pressures off upon the corresponding ordinates and connecting the points so located, the curve is formed as shown.

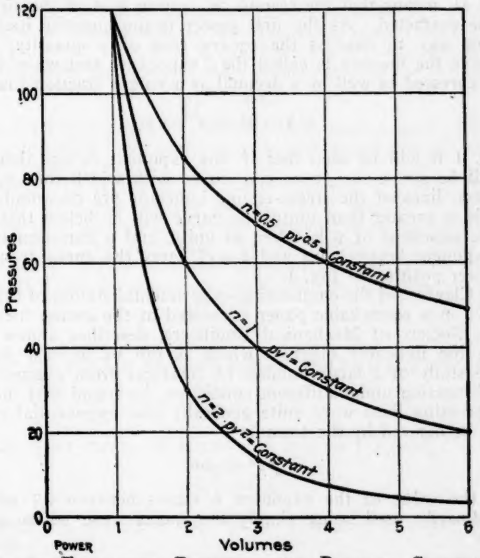


FIG. 1.—*pv* EQUATIONS PLOTTED WITH REGULAR CO-ORDINATES.

Suppose, however, that the variation of pressure with change of volume were such that the product of the pressure and the *square* of the volume were constant. The pressures for the several volumes would be as follows:

Vol.	Vol.^{2}		pressure		constant.
1	1	×	120	=	120
2	4	×	30	=	120
3	9	×	$13\frac{1}{3}$	=	120
4	16	×	$7\frac{1}{2}$	=	120
5	25	×	4.8	=	120
6	36	×	$3\frac{1}{3}$	=	120

Setting these values of the pressure off upon the several ordinates, the curve drawn through them is the lower one of Fig. 1.

Suppose, again, that the product of the pressure and the *square root* of the volume were constant, the pressures for the several volumes would be as follows:

Vol.	$\sqrt{\text{Vol.}}$		pressure		constant.
1	1	×	120	=	120
2	1.414	×	84.85	=	120
3	1.732	×	69.28	=	120
4	2	×	60	=	120
5	2.236	×	53.66	=	120
6	2.449	×	49	=	120

The expression 4^3 means that 4 is to be raised to the third power, or cubed, and that the second or square root of that power is to be extracted. Similarly $4^{\frac{1}{2}}$ means that the second or square root of the first power of 4 is to be extracted. As the first power is the quantity itself, this is a convenient way to express the square root of a quantity. The small figure above the number is called the "exponent" and when fractional it may be expressed as well by a decimal as a vulgar fraction thus:

$$\sqrt{4} = 4^{\frac{1}{2}} = 4^{0.5} = 2.$$

In Fig. 1 it will be seen that if this exponent is less than unity, the curve will be above the $pv = con.$ curve with which the expansion and compression lines of the steam-engine indicator are commonly compared, while if it is greater than unity, the curve will be below that in plotting which the exponent of v is taken as unity, and a variation of the value of the exponent between 0.5 and 2 will carry the curve from the upper to the lower position in Fig. 1.

J. Paul Clayton, of the engineering-experimental station of the University of Illinois, in a remarkable paper presented at the spring meeting of the American Society of Mechanical Engineers, describes a new method of analyzing the indicator diagram which is full of interest and promise. From the study of a large number of diagrams from engines of various types and running under different conditions, he found that the expansion and compression lines were quite generally true exponential curves, such as would be derived by the formula

$$pv^n = con.$$

but that the value of the exponent n varies between 0.7 and 1.34, the curve ordinarily used being simply the special case where $n = 1$ and $pv = con.$

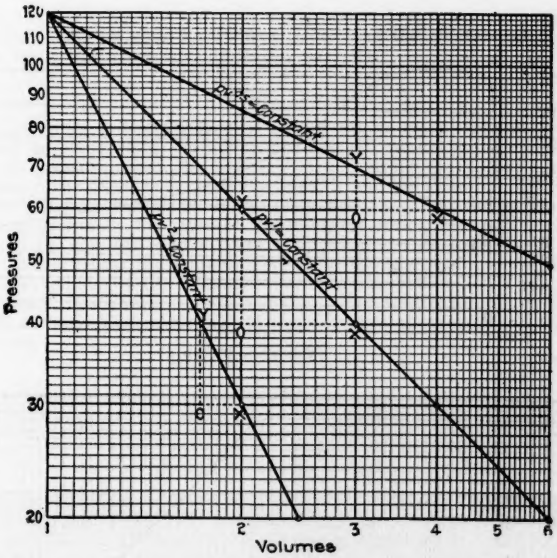


FIG. 2.—SAME EQUATIONS PLOTTED WITH LOGARITHMIC CO-ORDINATES.

If the ordinates, instead of being spaced equally as in Fig. 1, are spaced like the spaces on a slide-rule, so that the space from 1 to 2 equals the space from 2 to 4 or from 4 to 8, and the heights corresponding to the pressures are laid off in the same way, a curve having the formula

$$pv^n = con.$$

will become a straight line when laid out upon such a field. Paper ruled in this way and called logarithmic cross-section paper can be had from a dealer in draftsman's supplies.

A sample with the curves of Fig. 1 laid out upon it is shown in Fig. 2. The pitch or inclination of the curves (now become straight lines) depends upon their exponent, and the exponent may be determined by drawing a vertical and horizontal line intersecting each other and the curve as in Fig. 2, and dividing the length of the line OX by that of the line OY. Notice that for the middle curve, OX is the same as OY, and

$$\frac{OX}{OY} = 1.$$

On the lower curve OX is only one-half as long as OY, and

$$\frac{OY}{OX} = 2,$$

while for the upper curve OX is twice as long as OY, and

$$\frac{OY}{OX} = 0.5,$$

indicating thus the exponents used in the formulas by which the curves were laid out.

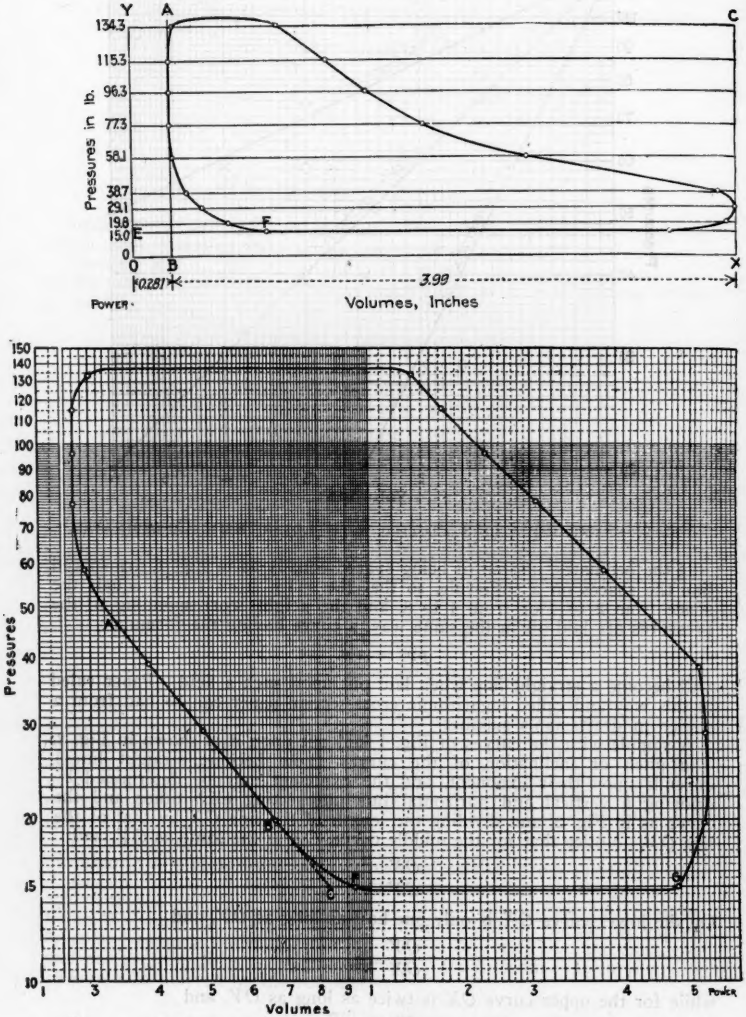


FIG. 3.—DIAGRAM FROM CORLISS ENGINE TRANSPOSED TO LOGARITHMIC SCALE.

Mr. Clayton plots not only the expansion and compression curves but the whole diagram upon logarithmic paper, and the diagram so produced offers many important advantages for analysis.

TRANSFERRING THE DIAGRAM.

Fig. 3 shows a diagram from a Corliss engine transferred in this way. It must be prepared by putting on the line OX, representing the absolute zero of pressures, and the line OY, representing the absolute zero of volumes. The distance between the line OX and the atmospheric line must be the barometric pressure at the place and time that the diagram was taken measured by the scale of the indicator spring used. The distance between the line OY and the line AB, marking the extreme left of the diagram, must bear the same proportion to the length AC of the diagram that the volume of the clearance bears to the piston displacement. In this engine the clearance is 7.04 per cent. The length AC of the diagram is 3.99 inches. The width YA of the clearance space on the diagram is therefore

$$0.0704 \times 3.99 = 0.281 \text{ inch.}$$

The barometer reading when the diagram was taken was 14.2 pounds per square inch and the scale of the spring used in the indicator was 79. The distance between the atmospheric and absolute zero lines is therefore

$$14.2 \div 79 = 0.18 \text{ inch.}$$

The logarithmic paper used consists of four 5-inch squares arranged two each way, as in Fig. 3, which shows the lower left-hand square complete with portions of the others. As the paper comes from the dealers it is numbered by heavy figures close to the ruling. These have been multiplied by 10 to give the scale of pressures as indicated by the figures at the left.

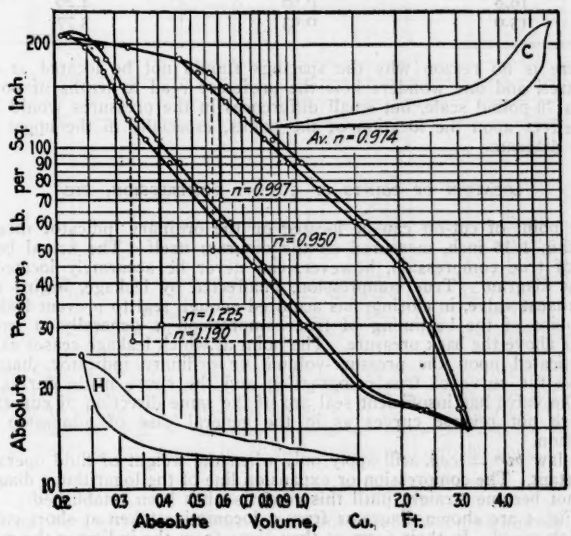


FIG. 4.—DIAGRAMS FROM LOCOMOTIVE USING SUPERHEATED STEAM.

The horizontal scale, that of volumes, will do as it is. The printed figures below the lower left-hand square will represent tenths of an inch and the intermediate rulings hundredths, except in the extreme left-hand division, 1 to 2, where there are 20 subdivisions of one-half a hundredth each. The figures in the right-hand square represent even inches.

A number of horizontal lines may be drawn across the diagram, as shown, and their intersections with the diagram transferred to the logarithmic paper. The point F, for instance, is at a height corresponding to 15 pounds, and its distance EF from the zero or clearance line is 0.93 inch. The point is thus located on the ordinate 0.93 on the scale of volumes (indicated by f on the horizontal scale) at the intersection of the horizontal line indicating 15 pounds. At the same height but at the horizontal distance 3.77 is located the point G.

In this way the several points of the indicator diagram are transferred to the logarithmic paper and lines drawn through them furnish the logarithmic diagram. The values of the pressures and volumes used in the diagram as given by Prof. Clayton are as follows:

Absolute pressure, pounds per square inch.	Distance from zero volume line of points on	
	Compression line.	Expansion line.
134.3	0.30	1.205
115.3	0.28	1.39
96.3	0.28	1.65
77.3	0.28	2.06
58.1	0.295	2.76
38.7	0.385	4.12
29.1	0.485	4.26
19.8	0.66	4.29
15.0	0.93	3.77

There is no reason why the spacings should not be located at even pressures, and one wonders how the professor read to tenths of pounds with a 79-pound scale, but small differences in the pressures would have little effect upon the location of the points, especially in the upper part of the diagram.

LOCATION OF POINTS OF CUT-OFF COMPRESSION, ETC.

The point of cut-off can be located on the ordinary indicator diagram to within 1/16 inch, measured on the diagram itself. The actual beginning of true compression, however, can never be accurately located on such a diagram. True compression, unaffected by leakage, begins after the exhaust valve, in closing, has acquired enough seal to prevent leakage. The point of the beginning of true compression is generally at least 5 pounds above the back pressure. The point at which leakage ceases cannot be indicated upon the pressure-volume or ordinary indicator diagram, because the curve of true compression and the curve made during the time the valve has insufficient seal are of the same direction of curvature, and are not reverse curves as in the general case of admission and expansion.

The law $pv^n = con.$ will apply only when the weight of fluid operating is constant. The compression or expansion line of the logarithmic diagram does not become straight until this condition has been established.

In Fig. 4 are shown diagrams from a locomotive taken at short cut-off and high speed. In their form as they came from the indicator the points of cut-off, release compression and admission are very difficult to locate.

but stand out sharply in the logarithmic diagram. Prof. Clayton says that these events, even though obscure in the pressure-volume diagram, may be located to well within about 1/16 inch on the logarithmic diagram, this length being equivalent to about 1/32 inch when retransferred to the *pv* diagram itself.

In most of the diagrams given in the paper the horizontal distances are taken as volumes in cubic feet, instead of lengths on the diagram. The author computes the volume of the cylinder for 1-inch length of diagram and multiplies the distance of the points from the line of zero volume by this factor. When the horizontal scale is used as diagram lengths the location of points determined as in the foregoing may be set off directly on the *pv* diagram at the intersection of the same pressure and volume lines, reversing the process by which Fig. 3 was plotted. When the horizontal scale is used for specific volumes, as in Fig. 4 and others from the paper, the reading for volume must be divided by the volume per inch of cylinder to obtain the horizontal distance on the *pv* diagram. The shape of the diagram will be the same whichever value is used for the horizontal scale; it will simply be shifted horizontally on the paper.

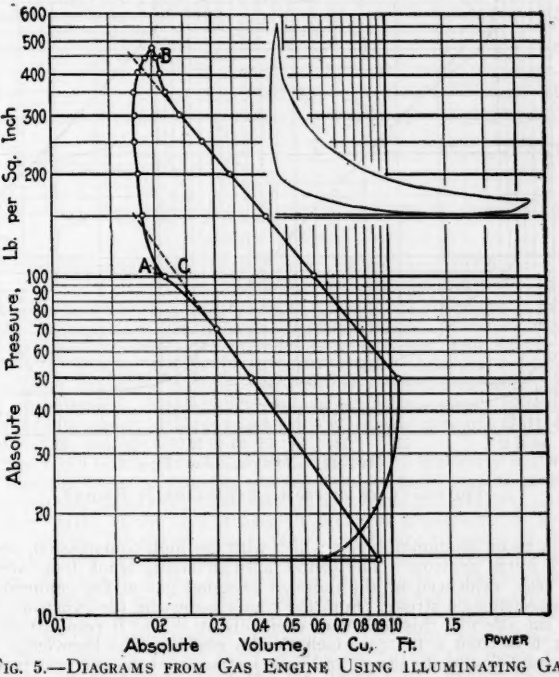


FIG. 5.—DIAGRAMS FROM GAS ENGINE USING ILLUMINATING GAS.

DETECTING LEAKAGE.

It has been pointed out that the expansion and compression curves will become straight lines in the logarithmic diagram only when the weight of fluid acting remains constant. They will not be straight lines, therefore,

when the quantity of fluid acting is being changed by leakage into or out of the cylinder. Except in the highly improbable case, where the leakage into just replenishes the leakage out of the cylinder, the effect of leakage will be to distort the expansion and compression curves so that they do not become straight lines on the logarithmic diagram.

Fig. 5 is a diagram from a 10 × 10-inch gas engine intended for producer

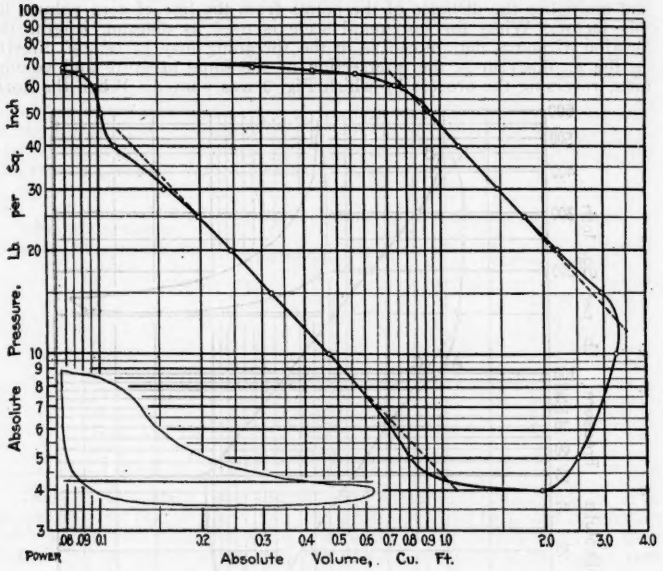


FIG. 6.—DIAGRAM FROM LEAKY CORLISS ENGINE.

gas but using illuminating gas, which with the high compression, produced a high initial pressure. The piston, a single-acting trunk type, allowed a large leak, evidenced by the noise of escaping gas at the commencement of the combustion stroke. Both the compression and the expansion curves show the effect of this leak very decidedly at *A* and *B* respectively.

Fig. 6 is from a 14 × 35-inch Corliss engine. The knowledge of the leaky condition of the piston and valves came from the engineer in charge. The expansion and compression lines indicate by their form at the upper ends a large leak from the cylinder, indicating a leaky piston or exhaust valve or both. The lines also show by the rising of the curves at the lower ends a considerable addition to the steam in the cylinder during expansion and compression. This steam could come only from a leak in the steam valve.

Fig. 7 is from an 11¼ × 22-inch double-acting ammonia compressor, the cylinder of which was known to be in very bad condition as regards wear and leakage of piston and valves. The reëxpansion curves, by the enormous amount of reëxpansion shown, indicate large leakage into the cylinder during this operation. The lower part of the compression curves, by rising, indicates leakage into the cylinder either past the piston

or through the discharge valves. The upper part of these curves indicates leakage from the cylinder either past the piston or through the suction valves.

Very smooth curves may be obtained in the indicator diagram even if there is considerable leakage taking place. The expansion and compression curves of the *pv* diagram, Fig. 6, are fairly regular, but the logarithmic diagram shows plainly that large leakage of two kinds was taking place during expansion and compression. Leakage which takes place during admission or exhaust has no effect upon the lines of the diagram as the weight of the medium is continually changing.

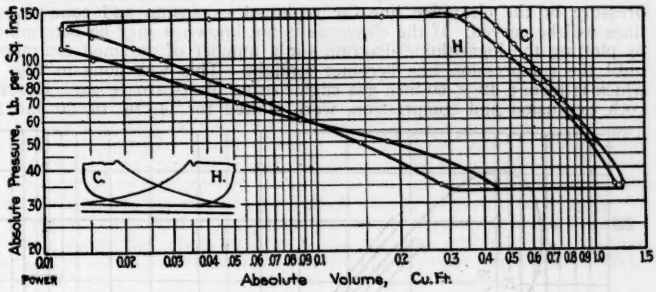


FIG. 7.—DIAGRAMS FROM LEAKY AMMONIA COMPRESSOR.

In discussing leakage it must be kept in mind that difference in pressure between two regions is the cause of the phenomenon. In the steam engine there are three pressures which must be considered, that in the steam chest, in the cylinder and in the exhaust passage. Leakage, being due to difference in pressure, becomes material only when this difference becomes considerable. The leakage into or out from a steam cylinder has been found to occur in most cases only when the pressure difference is over about 20 pounds.

In Fig. 6 the leakage into the cylinder, shown by the lower parts of the line, begins to occur at about 25 pounds absolute or 35 pounds lower than the pressure at admission. The leakage out from the cylinder, shown by the upper parts of the line, ceases to occur at a pressure of about 40 pounds absolute for the expansion curve and begins to occur at about 25 pounds absolute for the compression curve. The difference of pressure between the steam in the cylinder and that in the exhaust passage is about 35 pounds in the first case, and about 20 pounds in the second case.

This fact, founded upon many diagrams analyzed, permits the division of the expansion and compression lines roughly into three equal parts on the logarithmic diagram (when these lines extend from the initial pressure nearly to the back pressure): (a) the upper third, influenced by leaks *out from* the cylinder; (b) the middle third, practically *uninfluenced* by leakage; (c) the lower third, influenced by leakage *into* the cylinder. Thus fairly reliable values of n free from the effects of leakage may be obtained from the middle third of the lines. Ordinarily it is not possible to distinguish between two leaks occurring in the same third of the curves.

When only one kind of leakage exists it is possible to compute with fair accuracy the volume of leakage taking place during expansion or compression. In Fig. 5 the lines are extended (as shown dotted), giving the lines of constant weight of medium. If there had been no leak the volume at 100 pounds pressure would be that indicated by the intersection of the dotted extension of the compression line at *C*. The volume actually

present is indicated by the location of the point *A*. The difference between the volume indicated by *A* and *C* is a measure of the leakage. The important thing, however, is the indication that leakage is taking place so that it may be stopped.

DETERMINING THE CLEARANCE.

The curves of the indicator diagram will become straight lines upon the logarithmic paper only when the pressures and volume are measured from absolute zero. If there is any mistake made in locating the line of zero pressure or the clearance line the replotted expansion and compression lines will be curved. If the clearance is not known it may be determined by plotting the logarithmic diagram tor a number of assumed clearances until the compression line becomes straight. This was done in Fig. 8, which is from a 42 × 60-inch gas engine using blast-furnace gas. Curves, both expansion and compression, were plotted for 14, 16, 18, 20 and 22

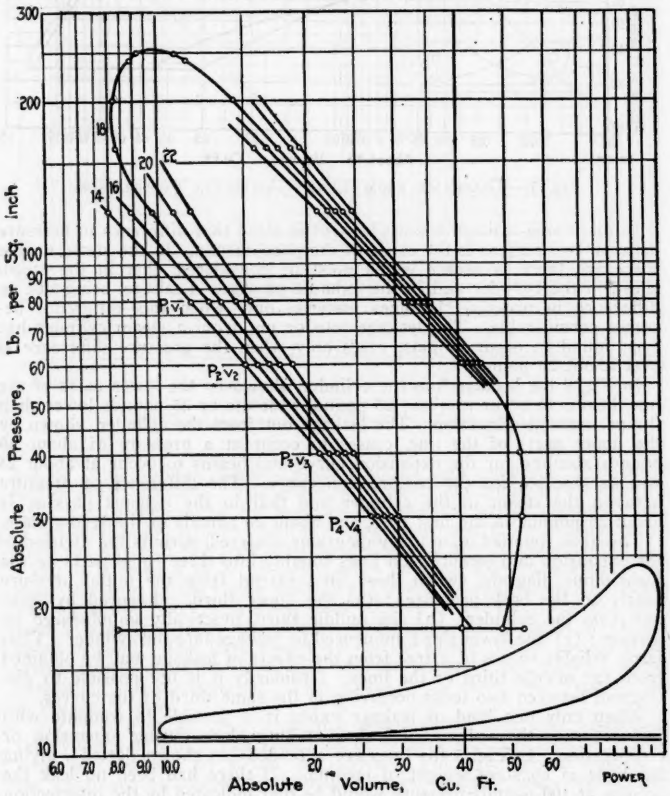


FIG. 8.—DIAGRAM FROM GAS ENGINE WITH BLAST-FURNACE GAS.

per cent. of clearance. With the true clearance of 18 per cent. the curves become straight lines. With values of the clearance less than 18 the curves become bent to the left; with the value above 18, to the right.

If the clearance is accurately known the position of the line of absolute zeros of pressure may be located in the same way, thus determining the barometric pressure if the atmospheric pressure and the scale of the spring are given, or the scale of the spring if the atmospheric pressure and reading of the barometer are given. The compression line gives naturally the better indication, as shown by the wider spacing and more decided curvature of the compression curves produced with the erroneously assumed clearances in Fig. 8.

STEAM CONSUMPTION FROM THE DIAGRAM.

The indicator diagram shows the pressure of the steam in the cylinder at the point of cut-off. From the steam tables its weight per cubic foot at this pressure may be determined. Knowing the volume of the cylinder up to cut-off and the number of times it is filled per hour, it would seem that the weight of steam thus metered off could be computed; and it could if the contents of the cylinder at the instant of cut-off were all steam. As a matter of fact, the steam consumption computed from the diagram in this way will be from 15 to 50 per cent. too low, because some of the entering steam has condensed upon the surfaces which have just been exposed to the cooler exhaust, and is present as water when cut-off takes place.

As the pressure is lowered by expansion below that corresponding to the temperature of this water, a portion of the steam is reevaporated, changing the course of the expansion line, and the most significant point brought out by Mr. Clayton's paper is that the line remains a true exponential curve and that the value of n bears a definite relation in any given cylinder to the proportion of the total weight of steam mixture which was present as steam at cut-off.

Furthermore, the relation of n to this initial quality for the same class of cylinder as regards jacketing has been found to be practically independent of engine speed and of cylinder size. If one can determine then the exponent of the curve of a given diagram, and knows the quality of the mixture at cut-off corresponding to that exponent, he can compute from the diagram the amount of steam actually used by the engine with a degree of accuracy limited only by the accuracy with which these quantities are known.

The method of determining the value of the exponent from the slope of the expansion curve has already been explained.

In Fig. 9 is given a chart from which the quality at cut-off for any value of n between 0.85 and 1.25 and for any pressure at cut-off between 40 and 160 pounds absolute may be measured. For $n = 0.95$, for instance, the quality at cut-off is a little less than 0.51; at 60 pounds a little less than half way between the curved lines representing 0.56 and 0.58. At 100 pounds it is nearly 0.62, etc.

This chart is plotted from the results of tests upon a 12 $\times$ 24-inch Corliss engine exhausting into a condenser in order that the condensate might be weighed, but at about atmospheric pressure. The speed had a slight effect upon the quality at cut-off and the chart is plotted for 120 r.p.m.

The effect of a change of pressure on the relations of the quality at cut-off and n is not great between the limits of 75 to 150 pounds. An approximate equation, correct for the conditions at 129 pounds absolute representing the average of the range mentioned, is

$$X_c = 1.245\ n - 0.576,$$

where

X_c = Proportion by weight of the mixture present as steam at cut-off; and

n = Exponent of v in the equation $pv^n = con.$

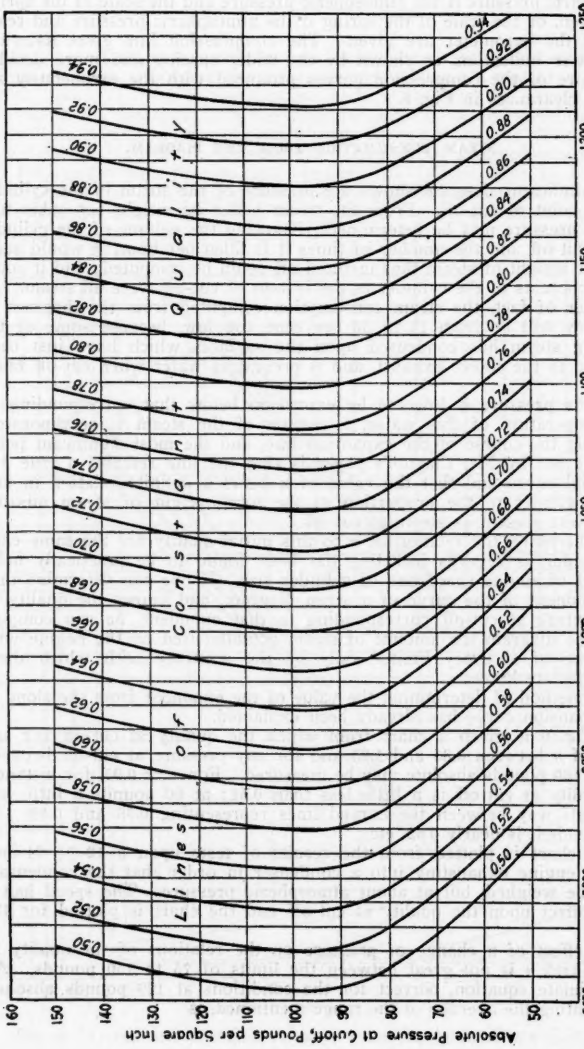


FIG. 9.—THE RELATION BETWEEN PRESSURE AT CUT-OFF AND VALUES n FOR CONSTANT QUALITY AT CUT-OFF.

This quantity for the ordinary case will be less than unity, and dividing the weight per cubic foot of dry saturated steam at the cut-off pressure, as given by the steam table, by it will give the weight per cubic foot of the mixture. Multiplying the volume of the cylinder up to the point of cut-off in cubic feet by the weight per cubic foot will give the weight of steam and water present in the cylinder at the point of cut-off.

But not all of this came in when the valve was opened for admission. There was some saved from the previous stroke by closing the exhaust valve before the stroke was finished. It has been demonstrated by George Duchesne ("Power," Jan. 10, 1911, p. 71) that this steam at the commencement of compression is dry saturated. After the exhaust valve has become fully seated and the weight of steam being compressed has become constant, the compression line will be straight upon the logarithmic diagram as between *A* and *B*, Fig. 3. If this line is projected downward as by the dotted line *BC*, the intersection with the back-pressure line of the diagram will indicate the volume of dry saturated steam of that pressure present. The weight of the volume at that pressure can then be taken from that present at cut-off and the difference will be the weight supplied per stroke, which multiplied by the number of strokes per hour will give the weight supplied from the boiler for that end of the cylinder.

The results of the applications made up to the present tend to show that the steam consumption of engines may be approximated from the indicator diagram to within an average difference of less than 4 per cent. from the test results. Individual examples, however, may show as much as 8 per cent. in rare cases. Further investigations of the relations of X_c and n with all types of engines running under all sorts of conditions may lead to the evolution of a set of tables or charts of these relations which can be used with confidence that they will give a sufficiently close approximation to the actual consumption of the engine to allow that consumption to be computed from the indicator diagram instead of laboriously being measured in an expensive test. An engine which would otherwise be tested perhaps once in its lifetime may have its water rate determined frequently with little more labor than is now spent upon the worthless uncorrected consumption from the diagram, and such easily-made determinations may serve all the purpose of expensive tests except where large bonuses or forfeits are involved, and even in these cases would serve as a check upon the test figures.

The paper contains tables of results from tests of numerous engines, an explanation of the development of the chart shown in Fig. 6, diagrams from various engines, including the Diesel, the Stumpf unidirectional flow, combined diagrams from compound and triple-expansion engines, and much material of value to the student of indicator diagrams.—"Power."

RECENT ADVANCES IN THE ART OF BATTLESHIP DESIGN.

By Naval Constructor D. W. Taylor, U. S. N.

In October, 1905, Great Britain laid the keel of a battleship materially larger than any before constructed and differing much from its immediate predecessors, notably in the fact that the heavy turret guns, instead of comprising two calibers, were all of the heaviest caliber—12 inches—and there was no intermediate battery of 6-inch caliber, the only caliber carried being 12 inches in the main battery and 3 inches in the secondary battery or torpedo-defense battery. This vessel, named the *Dreadnought*, being constructed with unprecedented rapidity and under circumstances of unusual and, for Great Britain, unprecedented secrecy, was, largely for this reason, the best advertised ship in the world. It has been the fashion

since to call large battleships Dreadnoughts, though in England, where the fashion originated, the expression super-Dreadnought is much used now, and doubtless we shall soon hear of super-super-Dreadnoughts.

While the *Dreadnought* herself has been thrown in the shade by the later vessels, some of which are more than fifty per cent. larger, she ushered in an era of world-wide competition in battleship building and rapid increase of size and power of individual ships.

Table I below shows the effective battleship tonnage on January 1, 1912, of the eight leading naval powers, divided between completed battleships of the pre-Dreadnought type and vessels of the Dreadnought type, built and building. For the purpose of this classification vessels having a main battery of all big guns, 11 inches or more in caliber, are classed as of the Dreadnought type. None of the eight nations of Table I is building battleships of any other type. Vessels over twenty years old are not included, so Table I shows approximately the battleship tonnage completed or laid down from 1891 to about 1906 as compared with that completed or laid down from about 1906.

TABLE I.

Effective Battleship Tonnage.

Nations.	Pre-Dreadnought type built. Tons of displacement.	Dreadnought type built and building. Tons of displacement.
Great Britain	617,500	483,350
Germany	252,712	359,120
United States	334,146	221,650
France	286,005	92,368
Japan	191,698	41,600
Russia	122,250	158,000
Italy	97,500	85,620
Austria	74,613	80,000

Table I brings out clearly the world-wide "speeding up" in battleship building of the last few years. The figures for Great Britain and Germany make it clear why some people in Great Britain consider that country to have made a colossal blunder when she forced the pace by building a new and more powerful type of battleship. The result, of course, is to relatively reduce in value earlier battleships, as to which we see, in the second column of Table I, England had a much greater superiority over Germany than indicated in the third column for battleships of Dreadnought type. Incidentally, Table I makes it clear why the United States is just yielding the place of the second naval power to Germany. Of completed battleships the United States even yet has more tons than Germany, but the latter country is building so many more tons that the United States will never regain second place unless there is a marked change of policy on the part of one country or of both.

Of course, the customary gaging of naval power by tonnage is not an exact method of determining fighting power, but there is no accepted method by which we can determine this except, perhaps, actual war. Even tonnage statistics are not strictly comparable. We know that we use two kinds of tons in this country, the short ton and the long ton. For measuring displacement of battleships there are virtually as many different kinds of tons in use as there are nations.

A freight steamer may be able to carry in cargo double the weight of her hull and machinery. Her displacement then may vary 200 per cent from her empty displacement. A battleship has a much larger proportion of fixed weight, but carries a large removable or variable load in her

coal, ammunition, stores and water. Such weights may amount to eighteen or twenty per cent. of the empty weight of a large battleship. It is the practice to include arbitrary amounts of consumable weights when fixing the legend displacement of a battleship, and, as the practice in this respect of the various nations varies, the designed or official displacement of a given battleship would be different in each country.

In most countries the designed displacement of a battleship corresponds to a condition in which she carries a rather small proportion of her capacity of consumable weights. That was also the practice in the United States until about three years ago, when a much larger proportion of consumable weights was included in the designed displacement. A 20,000-ton vessel under our previous practice, which was close to the average foreign practice, would be, say, a 21,500-ton vessel under our present practice.

There is another little peculiarity about warship tonnage as customarily stated. The displacement of a battleship is the designed displacement and never changes. We have never had much experience of vessels materially exceeding their designed displacement—I believe no vessel of war built on the designs of the Navy Department since 1903 has exceeded her designed displacement by a single ton—but I know of cases of vessels such as the *Dreadnought* herself which notoriously exceeded their design displacement when completed, yet are carried on all lists at the displacement of their original design.

Of course, the question of the proportion of consumable weights to be included in the official displacement of a battleship is largely one of expediency. Whatever the practice in this respect, the designer, in fixing position of armor and such matters, must, of course, consider all conditions of loading.

If the official displacement of a battleship is fixed rather light, she will make a slightly better speed on trial—the difference is very slight indeed for large battleships—and if existing tonnage is minimized, the chance of legislative authorization of more is improved. If, on the other hand, the official displacement is fixed rather heavy, since consumable weights which make her heavy are not included in cost of construction, the battleship will be cheaper per ton of displacement.

OFFENSIVE POWER.

A battleship, as you know, is a very complicated matter, a complete design being evolved by art as well as by science from many conflicting considerations. Perhaps the most salient characteristics bearing directly upon war efficiency are offensive power, speed and endurance, and defensive power.

There are many other essential characteristics, such as habitability, strength of structure, stability in intact and damaged condition, seaworthiness, etc., but for present purposes we must largely take these for granted.

In making Table I we adopt as the dividing line between the Dreadnought type and pre-Dreadnought types the characteristic of carrying a main battery of all big guns 11 inches or more in caliber. The Dreadnought was by no means a wholly novel type. The name has been borne in succession by a number of British men-of-war, and, curiously enough, when we compare the Dreadnought of 1905 with her immediate predecessor completed in 1875 (just thirty years before the Dreadnought of the present day was laid down) we find that the Dreadnought of 1875, like her successor, was, when completed, the largest, fastest, most powerful, and most heavily armored British battleship. Her main battery was uniform in caliber, consisting of four muzzle-loading rifles of 12.5 inches caliber, mounted in turrets. She carried also six rapid-fire guns—called then quick-firing—of 2½ inches caliber. The Dreadnought of 1905 carried a main battery uniform in caliber consisting of ten breech-loading

rifles of 12 inches caliber, mounted in turrets. She carried also twenty-seven rapid-firing guns of 3-inch caliber. Each of these Dreadnoughts had an armor belt extending from end to end; its thickness amidships was 11 inches in each case.

Compare them how we may, the twin-screw, reciprocating-engined, 14-knot, 10,800-ton Dreadnought of 1875 shows remarkable similarity of type to the four-screw, turbined, 18,000-ton, 21½-knot Dreadnought of thirty years later. One naturally asks how it happens that in 1905, when making a marked advance in battleships, there was recurrence to the type of 1875, particularly as regards battery. The reasons, I think, are two-fold. In the first place, in thirty years the process of evolution had nearly completed its cycle and the battery was approaching again the type of a generation before. There are on a battleship but two really satisfactory locations for turrets carrying heavy guns. One is forward of the machinery spaces in the center line of the ship, and the other is aft of the machinery spaces, also in the center line of the ship. So we find successors to the Dreadnought of 1875 in the English Navy mounting usually four heavy guns in these locations. But there was a steady evolution from the half dozen 2½-inch quick-firing guns of the old Dreadnought. The secondary battery grew into an auxiliary battery plus a secondary battery, so that in fifteen years, or about 1890, we find British battleships carrying four heavy or main-battery guns as before, but instead of a few puny quick-firing guns they carried ten or a dozen 6-inch guns, protected behind armor, in addition to smaller guns still.

These 6-inch guns, combined with four heavy 12-inch guns, remained the standard, one may say, for some ten years or so. In the King Edward class, designed about 1901, we find a change. Their immediate predecessors carried four 12-inch guns, twelve 6-inch guns behind armor and sixteen 3-inch guns unprotected. The King Edwards carried four 12-inch, four 9.2-inch in turrets, ten 6-inch behind armor and twelve 3-inch unprotected. This was the first appearance of the 9.2-inch gun upon the British battleship. The next British class—the Lord Nelsons—laid down in 1904, abandoned the 6-inch gun. The *Lord Nelson* carried four 12-inch, ten 9.2-inch in turrets, and fifteen 3-inch unprotected. Here, then, we have two sizes of heavy turret guns, and it is quite reasonable to suppose that in time the 9.2-inch guns would have grown larger until by natural evolution the all-big-gun one-caliber ship would have appeared.

But there was a second factor which accelerated the slow process of evolution. For thirty years, to my knowledge, it has been a truism of the ordnance officer that the gun is a weapon of precision. Indeed, the precision of heavy guns is astonishing. Twelve-inch shells fired from a modern high-powered gun and leaving the gun in exactly the same direction with exactly the same velocity may be expected in still air to strike a target 10,000 yards, or say 5⅔ miles, away within a very few feet of each other. But strangely enough, up to some ten years or so ago no navy appears to have realized the possibilities of the gun, or, at any rate, to have developed accuracy of shooting to an extent approaching anywhere near the inherent possibilities of precision of the gun. I think it may be justly claimed that the shooting of the United States Navy has always been as good as that of any other nation. In the Spanish-American war the shooting of the United States vessels was obviously far superior to that of the Spaniards, but, even so, the percentage of shots that hit was painfully small, and it is only the hits that count. But about the year 1900 there began a movement for improvement. It originated in the British Navy, but their first improvements were very soon adopted in the United States Navy, and both made rapid and remarkable progress. Telescopic sights were adopted and perfected. Formerly the gun sighter had to look simultaneously at his rear sight, his front sight, and the target. With the telescopic sight he has simply to look at the target. Cross wires in the

telescope which to the gun sighter seem to be in the plane of the target show the point of the target where the shot will strike if the sighting is correct. Methods were devised for increasing the rapidity of loading, and for facilitating training and elevating guns so that, regardless of the motion of the ship, the gun could be kept always pointed on the target and could be fired as fast as loaded. Guns, of course, were fitted with sight bars adjustable for the various ranges and also to allow for wind, etc., but it was soon found that there were variables present which no sight bar could take account of. For instance, after a gun had been warmed up by firing a number of shots it would not shoot quite the same as when cold. Two charges from different lots of powder would not shoot alike. Charges from the same lot of powder, if of different temperatures when loaded, would not shoot alike. Careful and systematic endeavors were made to eliminate or reduce to rule the above and other variables liable to produce erratic shooting, and it was found in the end that the most hits were made if the sight bar was corrected as necessary from observations of previous shots. Hence the development of observation stations as high up as possible, so that the "spotters," as they are called, can follow the fall of shell at extreme ranges. Elaborate systems of communication were also developed, so that the fire-control officers at the spotting stations could readily communicate necessary instructions to all guns.

Now it was found very early in the evolution of methods of fire control that the problem was very much complicated when it was attempted to handle simultaneously two different calibers of guns. Also, the largest guns were much more accurate at the longest ranges. Hence there arose a demand from gunnery officers for uniformity of caliber of heavy guns just about the time when, by a process of natural evolution, we were approaching this type. Accordingly we find the British *Dreadnought* and the American *Michigan*, which was designed very soon afterwards, carrying 12-inch guns and 3-inch guns only, the intermediate calibers, which were 7 inches and 8 inches in America and 6 inches and 9.2 inches in Great Britain, having disappeared.

It is interesting to note that at once the process of evolution again began. The small guns were called the torpedo-defense battery and were supposed to be of use only against torpedo vessels. But in England the 3-inch guns of the Dreadnought were replaced by 4-inch guns on her successors, and it is currently reported that the most recently laid down English battleship is to carry a torpedo-defense battery of 6-inch guns. In the United States the 3-inch guns of the *Michigan* were followed by the 5-inch guns of the *Delaware*, and on later vessels there is carried an improved type of 5-inch gun about as powerful as the 6-inch gun of ten years ago. Other nations which took up the Dreadnought type later never abandoned the 6-inch gun. Hence we may say that, broadly speaking, on the average the type of battery of today is again that of ten or fifteen years ago, consisting of a number of heavy guns in turrets and a number of 6-inch guns in broadside. The differences are, that on the much larger ships two or three times as many heavy guns are carried and the 6-inch guns are not so well protected, being regarded by many designers as useful against torpedo craft only. With the increase in size of torpedo craft and in range of the torpedo we may anticipate a demand for torpedo-defense guns which will put torpedo vessels out of action at longer ranges, and probably for larger calibers which are more accurate at the longer ranges. The art of fire control has not stood still, and the problem of the control of a mixed battery is not so difficult as it was some years ago. While the only thing that may be safely prophesied is that present types will be developed into others, it seems reasonably certain that the evolution of the torpedo-defense battery of today will be along one of two lines.

The torpedo-defense battery will be made larger in caliber, given more

protection, and relied upon for use in action against battleships, so that we will return to a mixed-battery type, or the main battery will be relied upon for torpedo-defense work and the torpedo-defense battery will disappear.

In concluding what I shall say about armament I would invite your attention to Fig. 1, indicating by skeleton deck plans the main battery distribution of the most recent battleships of the principal naval powers. Broadly speaking, the heavy turret guns monopolize the commanding positions and the small guns have to be given what is left. Most nations, the United States among them, locate the torpedo-defense guns below the heavy guns. They are thus better protected and easily supplied with ammunition, but are objectionably close to the water. Great Britain has hitherto located torpedo-defense guns at the level of and above the heavy guns. It is much more difficult to carry 6-inch guns thus than 4-inch guns, and if it is a fact that England has come to 6-inch torpedo-defense guns it is very probable that the change in caliber will be associated with a change in disposition.

As to the heavy guns, the disposition, as shown in Fig. 1, is most varied. I have already mentioned that the only satisfactory locations for turrets carrying heavy guns are forward and aft in the center line. Broadside fire is paramount under present conditions, as it is generally admitted that ships will fight broadside to broadside rather than end on.

A center-line mounting is the only one that permits a gun to be used with equal effect on each broadside. A gun mounted in the center line forward can also be used forward and one so mounted aft can be used aft. A gun mounted in the center line near the middle of the ship can be used, as a rule, for broadside fire only, and generally for rather a limited range only. Having in view the necessities of the machinery, it is difficult to provide satisfactory ammunition stowage for guns mounted near the middle of length of a ship.

As exemplified by the diagram of the *Michigan* in Fig. 1, it has been the practice in the United States Dreadnoughts to carry two heavy turrets at each end, thus having four turrets in the most satisfactory location. Additional turrets are located in the center line.

With two turrets at the end one must fire over the other. As this American arrangement has had the flattery of imitation by nearly all foreign nations, the history of its adoption may be of interest. It was first seriously proposed and advocated by the Bureau of Construction and Repair in 1895 in connection with the *Kearsarge* design. It was proposed then to locate an 8-inch turret behind and firing over a 13-inch lower turret. The gunnery experts of the day claimed that under such conditions the lower turret would be uninhabitable. The idea lingered, however, and about 1904-5 a thorough test was made upon the recommendation of Chief Constructor Capps. After preliminary experiments at the proving ground a crucial test was made on the monitor *Florida*. A 12-inch gun was removed from her turret and mounted so as to fire over it. Simple changes were made in the turret roof and the 12-inch gun was fired over it a number of times. The tests were progressive, the turret being occupied, I believe, first by four-footed animals, then by midshipmen, and so on up to rear admirals. With the turret top of proper thickness and tightly closed it was found perfectly feasible to occupy and work the lower turret when the upper turret is firing—a thing which will seldom be done, by the way—never in broadside firing.

Fig. 1 shows clearly that the latest battleships of nearly all nations have adopted this disposition.

It is seen from Fig. 1 that heavy turrets that are not placed near the ends in the center line are disposed according to three methods.

1. On the broadside firing on one side only.

2. On the broadside firing through a large arc on one side and a restricted arc on the other.

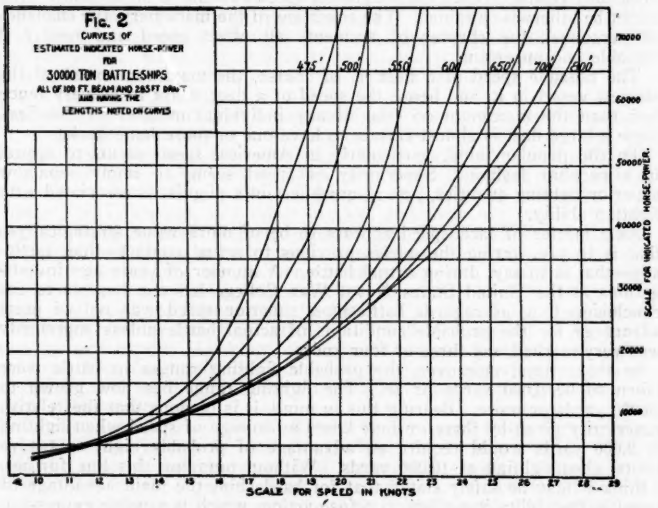
FIG. 1
BATTERIES OF BATTLESHIPS LAID DOWN IN 1905
AND SUBSEQUENT YEARS

YEAR OF LAYING KEEL	GREAT BRITAIN	GERMANY	UNITED STATES	JAPAN	FRANCE	ITALY	AUSTRIA	RUSSIA
1905	DREADNOUGHT 1 VESSEL 17 900 TONS 12 INCH GUNS 12 POUNDERS	HANNOVER CLASS 4 VESSELS 13 200 TONS 11 INCH GUNS 17 INCH GUNS	VERMONT CLASS 6 VESSELS 16 000 TONS 12 AND 8 INCH GUNS	SATSUMA CLASS 1 VESSELS 19 280 TONS 12 AND 12 INCH GUNS 6 INCH GUNS		SAN GIORGIO CLASS 4 VESSELS 9 830 TONS 12 INCH INCH GUNS 12 POUNDERS		
1906	BELLEROPHON CLASS 3 VESSELS 18 600 TONS 12 INCH GUNS 4 INCH GUNS	NASSAU CLASS 4 VESSELS 13 900 TONS 11 INCH GUNS 6 INCH GUNS	SOUTH CAROLINA CLASS 2 VESSELS 16 000 TONS 12 INCH GUNS 3 INCH GUNS					
1907	ST VINCENT CLASS 3 VESSELS 19 250 TONS 12 INCH GUNS 4 INCH GUNS		DELAWARE CLASS 2 VESSELS 20 000 TONS 12 INCH GUNS 2 INCH GUNS		DANTON CLASS 6 VESSELS 12 400 TONS 12 AND 9.4 INCH GUNS 12 POUNDERS		RADETSKY CLASS 3 VESSELS 14 500 TONS 12 AND 9.4 INCH GUNS 12 INCH GUNS	
1908		THURINGEN CLASS 4 VESSELS 21 000 TONS 12.2 INCH GUNS 8 INCH GUNS						
1909	NEPTUNE CLASS 3 VESSELS 20 200 TONS 12 INCH GUNS 4 INCH GUNS	KAISER CLASS 10 VESSELS 23 000 TONS 12.2 INCH GUNS 6 INCH GUNS	UTAH CLASS 2 VESSELS 21 825 TONS 12 INCH GUNS 2 INCH GUNS	KAWACHI CLASS 2 VESSELS 20 750 TONS 12 INCH GUNS 6 INCH GUNS		DANTE ALIGHIERI 1 VESSEL 19 000 TONS 12 INCH GUNS 17 INCH GUNS		CAIGOOT CLASS 4 VESSELS 23 300 TONS 12 INCH GUNS 47 INCH GUNS
1910	ORION CLASS 3 VESSELS 23 500 TONS 13.5 INCH GUNS 4 INCH GUNS		ARKANSAS CLASS 2 VESSELS 26 000 TONS 12 INCH GUNS 2 INCH GUNS		COURBET CLASS 6 VESSELS 23 467 TONS 12 INCH GUNS 22 INCH GUNS	CONTE DI CAVOUR CLASS 3 VESSELS 21 000 TONS 12 INCH GUNS 47 INCH GUNS	TEGETTHOF CLASS 4 VESSELS 20 000 TONS 12 INCH GUNS 4 INCH GUNS	
1911			TEXAS CLASS 2 VESSELS 27 000 TONS 14 INCH GUNS 2 INCH GUNS					

whereas if she were made 800 feet long the speed would be twenty-eight knots.

It is doubtful if sufficient length and space could be given to machinery in a 500-foot vessel to enable 70,000 horsepower to be developed; probably it would not be possible to drive such a vessel over twenty knots, owing to limitation of space for machinery.

However, assuming, for the present, that we could get 70,000 horsepower into each vessel, the weight of machinery would be approximately the same, regardless of the length of the vessel; but to build a 30,000-ton vessel 800 feet long would take a very much greater weight of hull than to build a 30,000-ton vessel 500 feet long. This additional weight of hull would have to come from the armor or armament, the 30,000 tons of displacement being fixed. Moreover, the thickness of armor protection for a given weight which could be placed on an 800-foot vessel would be very



much less than for the same weight applied to a 500-foot vessel.

It is evident, then, that the penalty paid for speed besides the direct weight necessary to provide for machinery, is the additional weight of hull necessary to provide a vessel of the length and form to enable it to be driven at the higher speed, and superposed upon this, the diminution of the armor thickness, or the restriction of the proportion of the area of the ship protected by armor resulting from the greater length to be protected. The solution of the problem of speed is then obviously a compromise between conflicting considerations, as is the case of so many other problems of warship design. For the latest United States battleships the designed speeds have varied from twenty and a half knots to twenty-one knots, trial speeds usually being a little better by half a knot or so.

It will be seen from Table II that these speeds are close to the average of those chosen by foreign nations.

The question of endurance of vessels of war, or the distance which they can steam with designed fuel capacity, is one which is very difficult to reduce to absolute rule.

In the first place, a battleship will seldom make the same run twice with

the same coal consumption. There are too many variable factors; the skill of personnel, condition of machinery, condition of bottom and weather conditions are all variables which enter into the question of endurance. There is hardly any quantity, moreover, which is so apt to be misrepresented as that of endurance, and so constantly exaggerated. A trial is made under most favorable conditions, the coal consumption being reduced to the minimum, and the endurance is stated to be that obtained by dividing the coal capacity by the consumption of the main engines, the trial being perhaps of a few hours' duration only. With such methods it is easy to obtain an endurance 50 or 100 per cent. greater than will be shown by the vessels in actual service under average conditions.

TABLE II.

DESIGNED SPEEDS AND ARMOR THICKNESS OF RECENT BATTLESHIPS OF VARIOUS NATIONS. (FROM JANE'S WARSHIPS.)

Nations.	Vessels.	No. in class.	Year of laying down first of class.	Designed speed.	Thickness of armor belt, maximum.
				Knots.	*Inches.*
Great Britain	*Monarch* Class	5	1910	21	12
	Orion Class	3	1909	21	12
	Colossus Class	2	1909	21	12
	St. Vincent Class	3	1907	21	9½
Germany	*Kaiser* Class	8	1909	20	11¾
	Thüringen Class	4	1908	20	10¾
	Nassau Class	4	1906	19.5	9½
United States.	*New York* Class	2	1911	21	12
	Arkansas Class	2	1910	20.5	11
	Utah Class	2	1909	20.75	11
Japan	*Kawachi* Class	2	1909	20	12
	Aki Class	1	1905	20.5	9
	Satsuma Class	1	1905	20	9
France	*Jean Bart* Class	6	1910	21	10¾
	Danton Class	6	1907	19.4	10
Italy	*Conte di Cavour* Class	3	1910	22.5	...
	Dante Alighieri Class	1	1909	23	...
Austria	*Tegetthof* Class	4	1910	21	11
	Radetzky Class	3	1907	20	9
Russia	*Gangoot* Class	4	1909	23	11
	Imperator Class	2	1903	18	8½

The matter of endurance has been brought more prominently to the front during the last few years by reason of the almost universal adoption of turbines for the propelling machinery of battleships. These have the unfortunate feature that if designed to give the best speed or the best results at or near the top speed, while giving better results for these conditions than reciprocating engines, they are very much less efficient than the reciprocating engines at ordinary cruising speeds. Now if the

greatest endurance is aimed at, it is necessary to steam at quite a low speed, below ten knots, in fact, but ordinarily endurances are figured on the basis of a ten-knot speed.

The net work done in driving a ship over a distance is proportional to the resistance and to the distance over which the resistance is overcome. For a constant distance the net work is proportional to the resistance, and if the resistance decreases indefinitely with speed the net work will also so decrease. But with any type of machinery the ratio between the net and gross work will decrease at low speeds as the speed decreases, so there will be some low speed at which the gross work done in steaming a given distance will be a minimum. In practice the speed for minimum gross work or maximum endurance is inconveniently small, but in most cases the endurance at the convenient speed of ten knots is almost as great as the maximum possible endurance.

The reduced economy of the turbines is associated not only with the economy of the turbine proper, but with the reduced efficiency of the type of propeller, which must be adopted to give the best all-round results for the turbine, hence the only fair basis of comparison is one involving all of the factors.

I have attempted to make such a comparison between the scouts *Birmingham, Chester* and *Salem,* tried two or three years ago. The *Birmingham* was fitted with reciprocating engines, and the *Chester* and *Salem* with turbine engines of different types. The curves of Fig. 3 show the

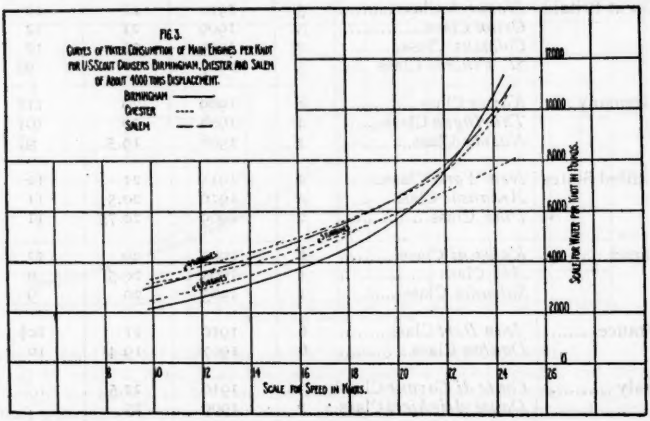


FIG. 3.
CURVES OF WATER CONSUMPTION OF MAIN ENGINES PER KNOT FOR U.S. SCOUT CRUISERS BIRMINGHAM, CHESTER AND SALEM OF ABOUT 9000 TONS DISPLACEMENT.
BIRMINGHAM
CHESTER
SALEM
SCALE FOR WATER PER KNOT IN POUNDS.
SCALE FOR SPEED IN KNOTS.

pounds of water used by the main turbines per knot for various speeds. The *Chester,* being fitted with various combinations, required three curves.

The basis of pounds of water used per knot was adopted in order to eliminate as nearly as possible the effect of type of boiler, efficiency of firing, etc.

I will invite attention to two facts brought out by Fig. 3. In the first place, the curves are still falling off at the speed of ten knots, so that if these vessels were to steam the maximum possible distance, time being not important, it would be policy, as already indicated, to adopt a speed even below ten knots.

In the second place, the *Birmingham,* with reciprocating engines, shows markedly better economy at the low speeds and worse at the high speeds.

Incidentally it may be remarked that the maximum trial speed of the *Birmingham* was materially below that of the two turbine vessels.

Although turbines have been adopted by practically all nations, including ourselves, for battleships, the superior economy of the reciprocating engines at cruising speeds caused us to return to them for the *Texas* and *New York*, now under construction, and contracted for a little over a year ago. Of our two battleships just contracted for, the *Nevada* and *Oklahoma*, one will have reciprocating engines and the other turbines arranged so as to promise better economy than hitherto.

Many devices have been proposed for obtaining the advantages of the turbines at high speeds and good economy at low speeds.

To gain the maximum steam economy for the turbine it should revolve much faster at high speeds than has been the practice, while the propeller of maximum economy should revolve much more slowly. When directly connected each hampers the other.

Among the methods to increase the economy of turbine-driven ships, I may note the following:

1. Gearing the turbine shaft to the propeller shaft. Then the turbine can run as fast as desired and the propeller shaft as slowly as desired within the limitations of the gearing. This method has been used abroad on merchant vessels with claims of success, and is being tried on a United States collier.

2. Indirect electrical drive—turbine of maximum economy driving electric generators, which in turn drive electric motors on the shafts.

This method is being tried on a United States collier under construction. It is heavier and more complicated than the simple gearing, but is more flexible than the gearing method and can be used for powers larger than would be undertaken with gearing at present.

3. Small reciprocating engines, to be clutched in at cruising speed and thrown out at high speeds, which exhaust to the turbines. This method is being tried on a United States torpedo-boat destroyer under construction, and shore tests of the apparatus indicate marked gain in economy.

As might be expected, each method tried has difficulties and objections peculiar to it, but steady progress is being made, and probably it will not be long before a satisfactory solution will be evolved. It appears to me that at present for vessels carrying the whole or a large proportion of their fuel supply in the form of oil, as do our most recent battleships, the ideal solution would be to fit Diesel engines, or the equivalent, driving generators, which in turn drive motors upon the shafts; this installation, however, to be of low power, adapted to drive the ship about ten knots only. For higher speeds turbines and boilers would be relied upon.

Such an arrangement would be slightly heavy as regards weight, but would have the advantage that the endurance would be doubled at cruising speed and an entirely independent means of propulsion would be provided, giving reasonable insurance against breakdowns. It would also be a step towards the generally anticipated use of the oil engine only for propulsion.

Such an installation would involve some difficult engineering problems, but I know of no reason why they could not be satisfactorily solved.

PROTECTION.

Let us now consider briefly the question of the protection of battleships.

In this connection we need to consider attack by gun fire, by torpedoes, by mines, and by explosives dropped from aeroplanes. The principal things to be protected are the buoyancy of the battleship, her stability, her vitals below water, such as engines, boilers, steam pipes, magazines, and steering gear; her armament; and the personnel directing the operation of the vessel.

As regards gun fire, we rely for protection almost entirely upon armor, whether vertical armor on the sides or horizontal or sloping armor in the

shape of a protective deck. It is evident that if the ship were composed of a very large number of small watertight compartments it would take a number of shots to destroy her buoyancy and stability, since each shot would reach but a limited number of the compartments. This principle of subdivision is relied upon to some small extent for protection against gun fire, but, as already stated, our main protection must be armor.

An ideal system of protection against gun fire would be one where the sides are covered with impenetrable armor from a point below the water line, as low as will ever be attacked by shell, to a point sufficiently high above the water line to insure that the ship would always retain her buoyancy and stability.

If, at this upper level, there were worked a level impenetrable deck we would have protection of buoyancy, stability and vitals.

To complete this conception we should have, rising from this upper level, impenetrable armor superstructures carrying impenetrable turrets, impenetrable conning towers, etc. Needless to say, this ideal is not attained in practice. The demand for offensive power and speed in battleships is so great that defensive power in many cases falls far short of the ideal and in no case actually reaches it.

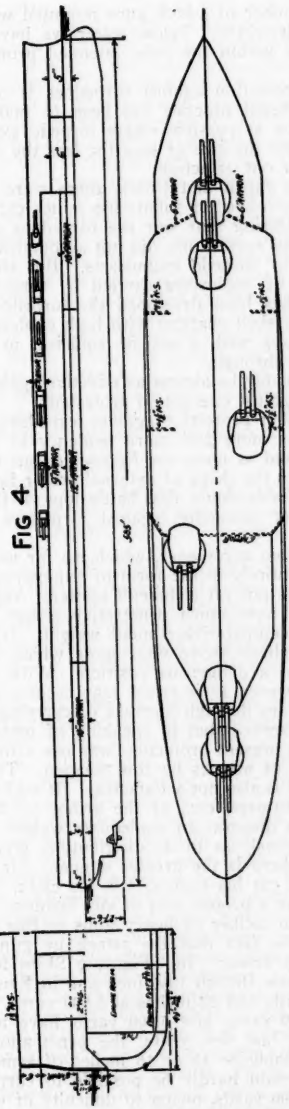
Full information as to actual armor protection of the battleships of the various nations is very difficult to obtain. Half a dozen battleships of the same size, carrying the same total weight of armor, would distribute it somewhat differently. However, as a rough gage of armor protection, we may use the maximum thickness of the main belt. This is given in Table II for a number of the most recent ships of various nations, the latest ship in each case coming first.

It will be observed that there is a tendency in nearly every nation to increase armor protection, judging by the maximum side-armor thickness given. Another fact noticeable from the table is the comparatively close agreement of a number of nations in the maximum side-armor thickness of their most recent ships. This is 12 inches in Great Britain, Japan and the United States, 11¾ inches in Germany, 11 inches in Austria and Russia, and 10¾ inches in France. There is no information as to the Italian ships, but, considering their speed, it is not likely that their armor is very heavy.

For many years there have been two opposing classes of thought as regards armor protection. On the one side we find the greatest importance given to the side armor with the idea of keeping the shell out of the ship as long as possible; on the other side we find great importance given to the horizontal armor, or sloping armor, the idea being that the shell would not do much damage provided it does not reach the vitals. Hence, we find material variations in the relative weights devoted to protective decks and side armor.

As illustrating current ideas of armor protection I invite attention to Fig. 4, giving the approximate distribution of armor upon some of the best protected of the most recent battleships; namely, the Argentine Republic battleships *Rivadavia* and *Moreno,* building in American shipyards upon American designs. It is seen that we have first a 12-inch belt extending over the midship portion of the vessel and tapering slightly forward and aft until we pass the last heavy-gun position, where it drops abruptly to a thickness of five or six inches. Above this belt is a uniform thickness of nine inches of armor extending to the upper deck and protecting the bases of turrets, smokestacks, engine hatches, etc. The barbettes and turrets rise above this level, their protection being about equivalent to that of the main belt. We also have conning towers projecting above this level.

The main protective deck, with a flat portion above the water line and sloping to the bottom of the side belt, is 2-inch nickel steel, and at a high level we have also 6-inch armor protecting the 12 to 6-inch guns. There



DISTRIBUTION OF ARMOR UPON ARGENTINA BATTLESHIPS.

are, in addition, a number of 4-inch guns mounted without protection on top of turrets and elsewhere. Below water we have a heavy bulkhead worked about 10 feet within the side, intended primarily for protection against torpedoes.

The question of protection against torpedoes is one which is by no means solved. The usual practice has been to make compartments as small and as numerous as possible where torpedo explosions were liable to occur, and the larger the size of ship the less the danger that a single torpedo would put her out of action.

In the war between Japan and Russia there were some very striking examples of the deadly effect of submarine mines carrying large charges of high explosives. During that war the torpedoes did not score many hits, and, when they did score a hit, did not accomplish the damage which had been anticipated by torpedo enthusiasts. But since then the speed, accuracy and weight of explosives carried by torpedoes have all been increased, and there has been developed the torpedo gun, or a torpedo carrying in a "gun" a shell charged with high explosives, which is fired, when the torpedo strikes, with a velocity sufficient to penetrate the ordinary ship through and through.

This device will probably be almost as effective against a ship protected with torpedo nets as against one not so protected.

The question of further protection against torpedoes has been talked of for years, and there is more and more tendency to fit such protection. It has usually been fitted as upon the *Rivadavia*, but there are advocates of fitting it externally in the shape of external armor far down on the ship.

There is little reasonable doubt that battleships of the near future will carry materially greater protection against torpedoes than those of the recent past.

As regards attack from aeroplanes, which, so far as can be anticipated at present, will come entirely in the form of explosives dropped from the aeroplanes, protection is not yet a difficult matter. Any bomb so dropped cannot be expected to have much penetrative power, and from present aeroplanes must have comparatively small weight. It would be possible to fit nets or light shelters above vital spots which would explode the bomb before it reached a dangerous position. With the rapid development of aeroplanes, however, their attack may become very serious within a comparatively few years through increase of carrying capacity.

It will have been observed that in speaking of protection against torpedoes I intimated that present protection was not satisfactory as regards the most recent forms of attacks by this weapon. The situation as regards attack by gun fire is also not satisfactory. It may be readily inferred from the varying thicknesses, etc., of the armor on the *Rivadavia* that the designer had at his disposal an inadequate weight of armor and has to ponder almost ceaselessly as to its distribution, giving, of course, the greater weight where there is the greater danger. He is in the position of the tailor who must cut his coat to suit his cloth, but finds his cloth quite inadequate to make a proper coat of any fashion.

The recent increase in caliber of heavy guns in this country and England has emphasized the fact that the attack by gun fire is markedly ahead of the defense by armor. In "Fighting Ships for 1911," by Jane, the penetration of the new British 13½-inch gun in Krupp armor is given as 26 inches at 3,000 yards and 22 inches at 5,000 yards.

While ranges of 3,000 yards and 5,000 yards have become very short for fighting within the last few years, the penetration of this gun at 10,000 yards would probably be 15 or 16 inches of armor, and, except in favorable weather, it would hardly be possible to carry on the fight at greater ranges than 10,000 yards, owing to difficulty of vision.

Bearing in mind that the heaviest armor carried by any British ship mounting 13½-inch guns is but 12 inches, we may say that these ships

can penetrate their own sides as far as it is possible to see. The same conclusions will apply to the 14-inch gun mounted on the American ships, and, indeed, we may say almost the same thing of the more powerful of the 12-inch guns whose use is practically universal.

As may be inferred from Table II, there is a tendency to increase armor thickness as the size of ships increases, and in the most recent United States ships this has been carried materially further than indicated in the table; but increases hitherto made can hardly be regarded as adequate, and it must be admitted that at the moment the gun is superior to the armor. Whether the armor will again forge ahead by superior combination of old elements, the development of still further improved armor, or the devotion to armor of a larger proportion of the displacement, it is impossible to say, but there is great need for improvement in protection, and it would seem fairly safe to prophesy that for some years to come we may expect to see the protection developed relatively more rapidly than the attack. I think that is certainly the proper and much-needed line of development. An alternative is to give up the fight and practically abandon armor.

In this connection I would invite attention to some extracts from a paper by Admiral Bacon before the British Institute of Naval Architects in the spring of 1910. He says:

"The problem of building a ship which can not be sunk by the explosion of a torpedo is one that has exercised the skill of naval architects, and the design of a ship which will not be incapacitated by such attack has hitherto baffled all solution.

* * * * * *

"As regards retention of the present thickness of armor protection, this is a matter which may before long undergo considerable modification, and the armor problem of the future appears to resolve itself into the answer to the following question:

"Supposing the guns of the enemy can with certainty pierce armor protection at reasonable fighting range, what is the most economical thickness of armor to adopt? Very many considerations are involved in the answer to the question, such as the position and thickness of horizontal armor, but, considering the enormous sacrifice in weight now made to carry thick armor protection, it is a matter that before long may undergo bold and radical revision."

Developments since the paper of Admiral Bacon appear to indicate that the bold and radical revision in armor protection has not come as yet, though now overdue. If the protection does not soon relatively increase, the only thing to do will be to abandon protection, just as the knights in the Middle Ages threw away their armor after gunpowder weapons were developed, and to evolve entirely novel types of ships. Personally, however, I think that the armor maker and the naval designer will not give up the fight and that protection will gain on attack.

In this connection, while slightly apart from my topic, I might say a word or two about the fast armored cruisers or "battle cruisers," now being built abroad. These vessels illustrate completely what I have said about the indirect effect of speed upon protection. They are as large as, or larger than, battleships, and their length is materially greater than that of the battleships, while their armor protection is very much less. They carry heavy guns as heavy as battleships, although not so numerous, and presumably would have to fight battleships at times. Their protection is so very slight, however, that against modern heavy guns they are practically on a par with the old protected cruisers, and the value of an enormous ship which will be put out of action immediately upon sighting the enemy appears at most problematic. They undoubtedly have their use for other purposes than the line of battle, such as scouting, but the value of any ship for scouting is likely to be largely reduced in the very near future by the development of aeroplanes.

The very name "battle cruiser" is a contradiction in terms, and the type is far from fixed. The weight which they carry in the form of protection is largely wasted, since it affords wholly inadequate protection, and if this type is further developed along logical lines we may expect to see a great increase of size with protection brought up to that of the battleship or an abandonment of protection along the lines forecasted by Admiral Bacon.—"Journal of the Franklin Institute."

THE STRUCTURAL COMPOSITION AND PHYSICAL PROPERTIES OF STEEL.

By Albert Sauveur,

Professor of Metallurgy and Metallography in Harvard University.

STRUCTURAL COMPOSITION OF SLOWLY COOLED STEEL.

Bearing in mind that hypo-eutectoid steel is composed of free ferrite and pearlite and that hyper-eutectoid steel consists of free cementite and pearlite, and knowing the proportion of carbon in pearlite (0.85 per cent.) and in cementite (6.67 per cent.), the structural composition of any steel may be readily calculated, provided we know the percentage of carbon it contains.

In case of hypo-eutectoid steel we have the following equations:

$$(1)\quad F + P = 100,$$

$$(2)\quad \frac{E}{100} P = C,$$

in which F represents the percentage of free ferrite in the steel, P the percentage of pearlite, E the percentage of carbon in pearlite, and C the percentage of carbon in the steel. The first equation expresses the fact that the steel is composed of ferrite and pearlite, and the second equation the fact that all the carbon in the steel is included in the pearlite. Assuming, for instance, that pearlite contains 0.85 per cent. carbon and the steel 0.50 per cent. carbon, the resolution of these two equations indicates that steel of that grade has the following structural composition:

$$F = \text{per cent. free ferrite} = 41.8.$$
$$P = \text{per cent. pearlite} = 58.2.$$

In case of hyper-eutectoid steel the following two equations may be written:

$$(1)\quad P + Cm = 100,$$

$$(2)\quad \frac{E}{100} P + \frac{6.67}{100} Cm = C,$$

in which P represents the percentage of pearlite, Cm the percentage of free cementite, E the percentage of carbon in pearlite, C the percentage of carbon in the steel. The first equation expresses the fact that hyper-eutectoid steel is composed of pearlite and free cementite, and the second the fact that the carbon in the steel is distributed between the pearlite and the free cementite, forming E per cent. of the pearlite and 6.67 per cent. of the cementite. Assuming the value of E to be 0.85 and the steel to contain 1.25 per cent. carbon, these equations give for a steel of that grade:

$$P = \text{per cent. pearlite} = 93;$$
$$Cm = \text{per cent. free cementite} = 7.$$

Supposing that pearlite or eutectoid steel contains 0.85 per cent. carbon, since the whole of that carbon is present in the cementite plates of pearlite, and since cementite contains 6.67 per cent. carbon (as called for by its chemical formula, Fe_3C), the percentage of cementite in pearlite may be readily calculated as follows:

$$\frac{6.67}{100} \times \text{per cent. cementite} = 0.85;$$

$$\text{hence, per cent. cementite} = 0.85 \times \frac{100}{6.67} = 12.74,$$

$$\text{and per cent. ferrite} = 100 - 12.74 = 87.26,$$

or, roughly, 1 part by weight of cementite to 6.6 parts by weight of ferrite.

If it be considered, however, (1) that the exact carbon content of pearlite is not, and hardly can be, known, (2) that it varies somewhat both with composition and treatment, and (3) that in commercial steel it is probably not far from 0.85 per cent., we are fully warranted to assume, for the sake of the great simplicity it introduces in the calculations, that pearlite contains exactly 1 part by weight of cementite to 7 parts by weight of ferrite, which would be the case if the eutectoid point corresponded to 0.834 per cent. carbon, as indicated below:

1 part cementite + 7 parts ferrite yields 8 parts pearlite,
or 12.50 per cent. cementite + 87.50 per cent. ferrite
= 100 per cent. pearlite;

and since cementite contains 6.67 per cent. carbon, 12.50 per cent. cementite will contain $6.67 \times .12.50 = 0.834$ per cent. carbon. Assuming, then, that such is the carbon content of eutectoid steel, so that 1 part of cementite gives exactly 8 parts by weight of pearlite, and noting that the carbon in the steel produces exactly 15 times its own weight of cementite,* the calculation of the structural composition of any steel becomes extremely simple.

In case of hypo-eutectoid steel (steel containing less than 0.834 per cent. carbon) we have

per cent. total cementite = per cent. total carbon × 15,
and per cent. pearlite = per cent. total cementite × 8;

or, more simply,

per cent. pearlite = per cent. carbon × 120;
i. e., $P = 120\,C$,

and, of course, per cent. ferrite $= F = 100 - P$.

With hyper-eutectoid steel (steel containing more than 0.834 per cent. carbon) the figuring is as follows:

Since 8 parts of pearlite contain 7 parts of ferrite and since in hyper-eutectoid steel the totality of the ferrite (total ferrite) is included in the pearlite (there being no free ferrite), we have

per cent. pearlite $= P = \frac{8}{7}$ total ferrite,
or, since total ferrite = 100 − total cementite,
$P = \frac{8}{7}$ (100 − total cementite).

But total cementite = carbon × 15, hence

$$P = \tfrac{8}{7}(100 - 15\,C), \text{ or,}$$

$$P = \frac{800 - 120\,C}{7},$$

and, of course, free cementite $= Cm = 100 - P$.

* This follows from the composition of Fe_3C indicated by the atomic weights of iron and carbon:
(3×56) Iron + 12 Carbon = 180 Fe_3C;
hence, one part carbon produces $\frac{180}{12} = .15$ parts Fe_3C, or cementite.

Summing up, in order to find the percentage of pearlite in hypo-eutectoid steel it will suffice to multiply its carbon content by 120 ($P = 120\ C$), the balance of the steel, consisting, of course, of free ferrite ($F = 100 - P$); to find the percentage of pearlite in hyper-eutectoid steel, the percentage of carbon in the steel should be substituted for C in the formula:

$$P = \frac{800 - 120\ C}{7},$$

and the balance of the steel will be made up of free cementite ($Cm = 100 - P$).

Taking, for instance, a steel containing 0.50 per cent. carbon, its structural composition will be:

$120 \times 0.50 = 60$ per cent. pearlite, and
$100 - 60 = 40$ per cent. ferrite.

If a steel contains 1.25 per cent. carbon the resulting percentage of pearlite will be

$$\frac{800 - 120 \times 1.25}{7},$$

or nearly 93 per cent., and the free cementite (Cm), 100 — 93 = 7 per cent.

CHEMICAL VS. STRUCTURAL COMPOSITION.

Disregarding the existence of impurities, the ultimate analysis of steel reveals the presence of so much carbon and so much iron. The proximate chemical analysis of steel reveals (in steel slowly cooled from a high temperature) the presence of so much iron and so much carbide of iron, Fe_3C. In a similar way we may consider two different structural compositions, an ultimate and a proximate one. The ultimate structural composition reveals the presence of so much total ferrite and so much total cementite, while the proximate structural composition informs us of the percentages of pearlite, free ferrite, and free cementite in the steel. It will be evident that the chemical proximate composition is identical to the ultimate structural composition, the names of the constituents only being different—iron and carbide in the first case, ferrite and cementite in the latter.

These various compositions are tabulated below:

		Constituents.
Chemical composition	ultimate: Fe	C
	proximate: Fe	Fe_3C
Structural composition	ultimate: total ferrite	total cementite
	proximate: pearlite	free ferrite, free cementite.

It is apparent that the proximate structural composition affords more valuable information than is obtainable through the other three kinds of analyses, for not only does it indicate the chemical nature of the proximate constituents but also their structural association and occurrence, upon which depend, to a very great extent, the physical properties of steel.

The structural composition, both ultimate and proximate, of slowly cooled steel is shown diagrammatically in Fig. 1, which will be readily understood. *ABC* represents the free ferrite in hypo-eutectoid steel, *ACD* the pearlite in hypo-eutectoid steel, *DCEF* the pearlite in hyper-eutectoid steel, *DFG* the free cementite in hyper-eutectoid steel, *ABEH* the total ferrite in any steel, *AHG* the total cementite in any steel, *ACEH* the pearlite-ferrite in any steel, and *AHFD* the pearlite cementite in any steel.

PHYSICAL PROPERTIES OF THE STRUCTURAL CONSTITUENTS OF STEEL.

It will be evident that the physical properties of commercial ferrite must resemble closely those of wrought iron and of very low-carbon steel. Ferrite, therefore, is very soft, very ductile, and relatively weak, having a

ductility corresponding to an elongation of at least 40 per cent. and a tensile strength of some 50,000 pounds per square inch. It is magnetic, has a high electric conductivity, and is deprived of hardening power, industrially speaking at least, since carbonless iron cannot be materially hardened by rapid cooling from a high temperature.

The properties of pearlite are evidently those of eutectoid steel in its normal—*i.e.*, pearlitic—condition, from which we may infer that pearlite has a tenacity of some 125,000 pounds per square inch, an elongation of some 10 per cent., that it is hard, and that it possesses maximum hardening power.

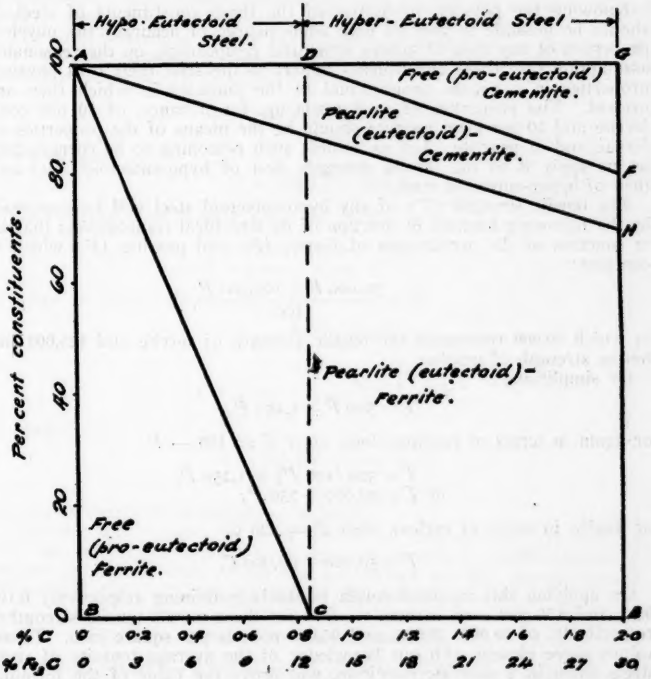


FIG. 1.

With the exception of its very great hardness, little is positively known as to the physical properties of cementite. It may be assumed, however, that so hard and brittle a substance must greatly lack tenacity. Its tensile strength probably does not exceed 5,000 pounds per square inch, and may be considerably less, while its ductility must be practically nil. It possesses no hardening power.

These properties of the constituents of steel in its normal condition are tabulated below:

Constituents.	Tensile strength, pounds per sq. in.	Elongation, per ct. in 2 in.	Hardness.	Hardening power.
Ferrite	50,000 ±	40 ±	Soft.	None.
Pearlite.....	125,000 ±	10 ±	Hard.	Maximum.
Cementite.....	5,000 (?)	0	Very hard.	None.

TENACITY OF STEEL VS. ITS STRUCTURAL COMPOSITION.

Knowing the physical properties of the three constituents of steel, it should be possible to foretell with some degree of accuracy the physical properties of any steel of known structural composition, on the reasonable assumption that these constituents impart to the steel their own physical properties in a degree proportional to the amounts in which they are present. The properties of steel made up, for instance, of 50 per cent. ferrite and 50 per cent. pearlite should be the means of the properties of ferrite and of pearlite. Let us assume such reasoning to be correct, and let us apply it to the tensile strength, first of hypo-eutectoid steel and then of hyper-eutectoid steel.

The tensile strength (T) of any hypo-eutectoid steel will be expressed by the following formula in function of its structural composition; that is, in function of the percentages of ferrite (F) and pearlite (P) which it contains:

$$T = \frac{50{,}000\,F \times 125{,}000\,P}{100},$$

in which 50,000 represents the tensile strength of ferrite and 125,000 the tensile strength of pearlite.

Or simplifying:

$$T = 500\,F + 1{,}250\,P;$$

or again in terms of pearlite alone, since $F = 100 - P$

$$T = 500\,(100\,P) + 1{,}250\,P,$$
$$\text{or } T = 50{,}000 + 750\,P;$$

or finally, in terms of carbon, since $P = 120\,C$

$$T = 50{,}000 + 90{,}000\,C.$$

On applying this simple formula to steels containing respectively 0.10, 0.25, and 0.50 per cent. carbon, we find for these metals tensile strengths, respectively, of 59,000, 72,500, and 95,000 pounds per square inch. These values agree closely with our knowledge of the average tenacity of such steels when in a pearlitic condition, and prove the value of the formula derived from the considerations outlined above as to the relation existing between the physical properties of steel and its structural composition. It should be borne in mind that in working out this formula it has been assumed that pearlite contains 0.834 per cent. carbon.

The values obtained for various eutectoid steels should be accurate only for steel in its pearlitic condition. It should be noted, however, that steels forged and finished at a fairly high temperature are practically in this condition, so that the formula may be used, and fair results expected, to calculate the tensile strength of such hot-forged steels. If the steel be forged until its temperature is quite low, and especially if it be cold worked, it is well known that its tensile strength is generally increased. Neither can the formula be used, of course, in the case of hardened steel or of steel castings. It may, however, be applied to steel castings which

have been properly annealed, when the tensile strength may be brought up to the level of steel forgings finished fairly hot.

Again, the formula is of value only in case of commercial steels containing the usual proportions of impurities, especially of manganese. It applies only to steels in which the percentage of manganese varies roughly with the carbon content from some 0.20 to 0.80 per cent. The presence of a larger proportion of manganese would increase the tenacity materially.

Passing to the tensile strength of hyper-eutectoid steel, our ignorance as to the tenacity of cementite does not permit the writing of a formula with the same degree of confidence. Let us assume, tentatively, however, that cementite has a tensile strength of 5,000 pounds per square inch, and then proceed as we did in the case of hypo-eutectoid steel.

The tensile strength of any hyper-eutectoid steel may be expressed by the following formula in terms of the percentages of pearlite (P) and cementite (Cm) which it contains:

$$T = \frac{125{,}000\,P + 5{,}000\,Cm}{100},$$

or simplifying

$$T = 1{,}250\,P + 50\,Cm,$$

or, in terms of pearlite only, since $Cm = 100 - P$,

$$T = 1{,}250\,P + 100\,(50 - P)$$
$$T = 5{,}000 + 1{,}200\,P,$$

or since, as previously shown,

$$P = \frac{800 - 120\,C}{7}$$

$$T = 5{,}000 + 1{,}200\,\frac{800 - 120\,C}{7},$$

or simplifying

$$T = \frac{995{,}000 - 144{,}000\,C}{7},$$

or approximately $T = 142{,}000 - 20{,}600\,C$.

Applying this formula to steels containing, respectively, 1.25 and 1.59 per cent. carbon, we find for their respective tensile strengths 116,250 and 111,100 per square inch, which are fair values for the average tenacity of pearlitic steels of those degrees for carburization.

STEEL OF MAXIMUM STRENGTH.

From the preceding considerations it seems evident that eutectoid steel must possess maximum tensile strength, since the influence of the presence of ever so small an amount of free ferrite in hypo-eutectoid steel or of free cementite in hyper-eutectoid steel must necessarily be a weakening one, because of the relative weakness of free ferrite and free cementite as compared to the strength of pearlite. By most writers, on the other hand, steel of maximum tenacity is often stated to contain in the vicinity of 1 per cent. carbon—that is, to be slightly hyper-eutectoid.

It is not clear, however, that the results upon which the statement is based were obtained in testing steel in its *pearlitic condition.* On the contrary, it seems probable that a large number of the steels tested were in a sorbitic rather than in a pearlitic condition because of relatively quick cooling through the critical range. And, while it appears that pearlitic steel must have its maximum tenacity when composed entirely of pearlite, it may well be that when in a sorbitic condition maximum strength corre-

sponds to a higher degree of carburization—*i.e.*, 1 per cent.—because sorbite may contain, and indeed often does contain, more carbon than pearlite. Indeed, the cases on record show that when the steels were made pearlitic through very slow cooling maximum tenacity corresponds closely to the eutectoid composition. Arnold, for instance, tested a series of very pure carbon steel and after slow cooling in the furnace from 1,000° C. he found a very sharp maximum in the tenacity corresponding to 0.89 per cent. carbon. On cooling these same steels in air, on the contrary, and therefore making them sorbitic, maximum tenacity corresponded to 1.20 per cent. carbon. Harbord likewise ascertained the tenacity of very pure steels, and found that after slow cooling (in the furnace) from 900° C. the maximum tenacity corresponded to 0.947 per cent. carbon.

DUCTILITY OF STEEL VS. ITS STRUCTURAL COMPOSITION.

From the known ductility, as expressed by its elongation under tension, of ferrite and the known elongation of pearlite, respectively 40 and 10 per cent. in two inches, it should be possible to work out a formula expressing the ductility of any hypo-eutectoid steel in the annealed (pearlitic) condition. In terms of ferrite and pearlite the ductility should be

$$D = \frac{40\,F + 10\,P}{100},$$

or simplifying

$$D = .4\,F \times .1\,P,$$

or, in terms of pearlite alone, since $F = 100 - P$,

$$D = .4\,(100\,P) \times .1\,P = 40 - .3\,P;$$

and since $P = 120\,C$, the ductility in terms of carbon will be

$$D = 40 - 36\,C.$$

Pearlitic steels, for instance, containing 0.25 and 0.50 per cent. carbon should have elongations, respectively, of 31 and 22 per cent.

REDUCTION OF AREA VS. STRUCTURAL COMPOSITION OF STEEL.

In a similar way we may calculate the reduction of area of any slowly cooled (*i.e.*, pearlitic) hypo-eutectoid steel, on the assumption that pearlite has a reduction of area of 15 per cent. and ferrite a reduction of 60 per cent. The reduction of any steel will then be, in terms of ferrite and pearlite:

$$R = \frac{60\,F + 15\,P}{100} = .60\,F + .15\,P,$$

or, in terms of pearlite alone, since $F = 100 - P$,

$$R = .60\,(100 - P) + .15\,P = 60 - .45\,P;$$

or, finally, in terms of carbon, since $P = 120\,C$,

$$R = 60 - 54\,C.$$

Pearlitic steels, for instance, containing 0.25 and 0.50 per cent. carbon should have reductions of area, respectively, of 46.5 and 33 per cent.—"Journal of the Franklin Institute."

THE INFLUENCE OF CARBON ON THE CORRODIBILITY OF IRON.*

By C. Chappell.

1. *Influence of Carbon on Corrodibility.*—(*a*) In rolled, normalized, and annealed steels, the corrodibility rises with carbon content to a maximum at saturation point (0.89 per cent. carbon), and falls with further increase of carbon beyond this point. (*b*) In quenched and tempered steels a continuous rise in corrodibility occurs, with increase of carbon within the range investigated (up to 0.96 per cent. carbon), no maximum corrodibility at saturation point being found in these steels.

2. *Influence of Treatment on Corrodibility.*—Quenching increases the corrodibility to a maximum; annealing tends to reduce it to a minimum; whilst normalizing gives intermediate values. The influence of tempering varies with the tempering temperature.

3. *Factors Determining Corrodibility.*—The electro-motive forces between the pearlite and ferrite, and between the components of the pearlite itself, are the principal factors determining the corrodibility of unsaturated pearlitic steels above 0.4 per cent. carbon. In mild structural steels this galvanic action, due to differences of potential between the constituents, is accompanied by galvanic action between the ferrite crystals themselves. These differences of electro-potential between the ferrite crystals are the result of differences in their orientation. The state of division of the pearlite, and the presence of internal stresses in the steel, may also exert a considerable modifying influence on the foregoing factors.

4. The influence of time on the rate of corrosion varies with different steels. In a low-carbon steel it is shown to be practically directly proportional to the length of immersion.

5. The influence of corrosion on the resistance offered by a low-carbon steel to alternating stress is not appreciable within a period of five months' immersion.

6. Three per cent. of tungsten produces practically no change in the corrodibility of carbon steels.

7. Decarbonization increases the resistance to corrosion.

8. The two oxides, FeO and Fe_3O_4 (mill scale), both exert a microscopical pitting effect on steel when in contact with it in sea water.

THE SOLIGNAC-GRILLE BOILER.

A WATER-TUBE BOILER IN WHICH THE STEAM IN FORMATION DOES NOT PASS THROUGH THE WATER.—THE ENTIRE BOILER MAY BE TURNED ON A PIVOT OUT OF THE SETTING.

Among boilers employing small-diameter water tubes for rapid steaming the new Solignac-Grille boiler, made by Grille & Co., Paris, France, presents some original features in its circulating and superheating systems and the means employed for cleaning the tubes. The manner of circulating the water and passing the steam through the tubes is quite different from that employed by other boilers. One novelty is the introduction at the feed-water end of the tube of a supplementary resistance, preventing escape of steam in a direction to interrupt the inflow of water and admitting water only as fast as it can be vaporized in the tube.

Before describing the boiler it might be well to briefly outline the experimental demonstration of the reasons for adopting a device to insure

*Abstract from a paper read before the Iron and Steel Institute, in London, May 9, 1912.

regular circulation, which were given at some length by "Engineering" in an abstract of a paper read by G. Hart before the Institution of Naval Architects.

Investigations on water circulation in boilers by Mr. Bellens have shown that the evolution of steam is not continuous, but intermittent and in the form of very large steam bubbles. By means of a sighting hole fitted with a glass cover it was also shown that the water descends between two escapes of steam by the path of this escape. It was to do away with such spasmodic emissions of steam and reverses of direction that Mr. Solignac conceived the idea of fitting at the lower end of the tube a diaphragm creating a supplementary resistance which should prevent any escape of steam at the lower end.

As a basis for this design he took the result of the following experiments: Two similar vessels, such as are shown in Fig. 1, were placed in the same horizontal plane and connected by a horizontal metallic tube. The tube was heated and the steam escaped intermittently from the two ends. The heated portion became red hot and the water which the tube contained was converted into steam. This steam escaped as often as its pressure exceeded the resistance caused by the head of water above the tube; then it escaped with a sudden rush and each time water again filled the entire tube. When the water rushed into the tube, a rapid hammering could be heard, which decreased as the temperature of the feed water rose.

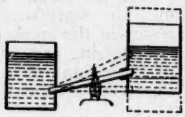


FIG. 1.

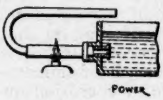


FIG. 2.

By carefully raising one of the vessels so as to give more inclination to the tube, the phenomena were modified according to the inclination. As long as it was slight and did not exceed the diameter of the tube the effects observed were nearly the same as when the tube was horizontal, but as the inclination of the tube increased a larger quantity of steam escaped at the side next to the upper end, while the water in the vessel on this side was also heated to a higher temperature.

That the escape of steam was never regular, and had periods of variable intensity, was shown by hanging on the upper end of the tube a small thin shutter, which lifted with each discharge of steam. While there was only an occasional escape of steam from the lower end of the tube a liquid film of viscous appearance could be seen descending in it just prior to each discharge of steam at the upper end. With the same apparatus a reducing diaphragm was fitted at the lower end of the tube and the phenomena of circulation became quite different. Steam then escaped continuously from the free end. The tube no longer became red hot, the water in the vessel on the diaphragm side remained cold and the water in the other vessel was heated rapidly. Finally, the level of the water became lower in the vessel on the diaphragm side and rose in the other an amount varying with the inclination of the tube. This experiment was supplemented by another with the bent tube shown in Fig. 2. The same phenomena were observed and the same results obtained, indicating that the diaphragm is the proper regulating device to afford a continuous flow of steam.

The sectional view, Fig. 3, reveals the general design and arrangement of the boiler, consisting of a horizontal drum with two attached compartments into which the tubes are expanded. Water is admitted into

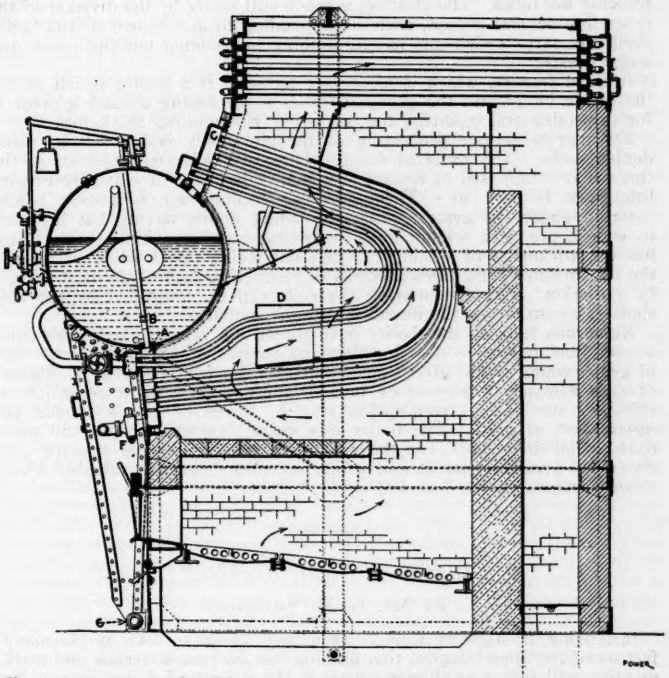


FIG. 3.—SECTIONAL VIEW THROUGH GRILLE BOILER, EQUIPPED WITH SUPERHEATER AND ECONOMIZER.

the lower compartment *A* through a valve *B* and its flow into the tubes is restricted by the diaphragm previously mentioned. Steam escapes from the tubes into the upper compartment or steam chest *C* and passes into the main drum from which the supply of water is drawn. The steam, not being compelled to cross the water surface but escaping above it, is much drier than in boilers where it has to pass through the water. The proportion of moisture is very small, not exceeding a few thousandths. The arrangement does away with tumultuous ebullition and consequent rushes of water, particularly as the main drum and steam chest are not exposed to intense heat, as may be seen from the path of the gases of combustion, indicated by the arrows in the illustration. The superheating tubes are shown at *D* and may be made auxiliary to the regular heating surface by flooding them through the valve *E*.

Next to the diaphragms the manner of cleaning is most interesting. The valve *B* is closed, shutting off the main drum from the lower compartment. Then by opening valve *F* steam under boiler pressure from the upper compartment flows in a reverse direction through the tubes and blows out the scale. The tubes, now containing no water, become hotter than normally and, expanding, loosen the scale which is readily carried away through the blow-off valve. The tubes are only left 30 seconds without water, which is long enough to detach the scale without

injuring the tubes. The cleaning is made still easier by the division of the tubes into several groups, each corresponding to an element of the boiler shell, the various elements having nothing in common but the steam and water chests.

A third feature, which is altogether unusual, is a device which allows the whole boiler and the group of tubes to be swung around a pivot *G* for inspecting and repairing the brickwork surrounding the boiler.

For marine service preheating of the air supply is possible by using double walls. The external facing plates of the boiler opposite to the furnace are composed of riveted sectional cells inside of which refractory brickwork is built up. Through these sections air is passed before entering under the grate so that the ashpit doors, having but little air to supply, are very seldom used in ordinary service. This arrangement has the advantage of cooling the external facing plates while preheating the air, and this constitutes a means of recovering part of the heat wasted by radiation. The air entering the cells can be controlled by movable shutters according to the intended rate of combustion.

Numerous tests on this boiler have shown the rate of combustion with a maximum output to average about 30 pounds of coal per square foot of grate, which would give an evaporation from feed water at 55 degrees of approximately 9 pounds of water. With a very slight reduction of efficiency the rate of combustion may be increased to 40 pounds per square foot of grate. The boiler is a quick steamer, occupies comparatively small floor space, has the advantage of easy internal cleaning and, above all, produces dry steam, which becomes especially valuable where steam turbines are used aboard ship.—" Power."

DIESEL ENGINE DESIGN.*

By Mr. D. M. Shannon.

In passing, it might be well to warn those about to take up the manufacture of the Diesel engine, that nothing but the best materials and workmanship will suffice. Otherwise there is the danger of doing a promising motor great harm in its infancy. These are points that are usually brought home very severely to steam engineers on first taking up the oil engine. The reason for greater accuracy and better material is, of course, due, in the first instance, to the high pressures that have to be dealt with, and, secondly, to the small volume occupied by the power medium. In an oil engine one cubic inch of oil for power purposes is, on an average, equivalent to 5,500 cubic inches of steam used in a steam engine. Hence, the measuring and burning arrangements must be made with great accuracy; any slight variations from which might mean large variations in power and efficiency of the machine. Experience has shown that only a few thousandths of an inch here and there makes the difference between a good running motor and complete failure.

The oil engine, it should be remembered, is still in its infancy, and the greatest care is required in its design and manufacture. Oil-engine workers recognize that the piston engine is only a temporary solution to the problem, and that a reversion to the turbine principle will be the natural outcome of the inherent limitations and disadvantages of the reciprocating form of motor.

Great efforts are being made with engines of the turbine form, and only those engaged in the industry know how enormous are the difficulties to be overcome. It is interesting to note that the only reason why oil engines have been practicable for so many years is the fact that, the

* Extract of a paper read before the Institution of Engineers and Shipbuilders.

surfaces coming into contact with the flame receive a little rest between the explosions in which to dissipate the heat.

If the process be made continuous, as in the turbine form of motor, then trouble from this source might be overcome by allowing one set of blades to cool down while another set carried on the work, or by trying a turbine of the bladeless form. Again, the steam turbine only shows superiority over the reciprocating engine when it is made condensing, and this advantage cannot be gained in the internal-combustion turbine over the reciprocating engine. There is also the difficulty of obtaining an efficient compressor of the turbine form, and it is not logical to carry out one part of the process with a piston machine and the other part with a turbine. This difficulty would be minimized with a producer which would produce gas at a pressure of, say, 150 pounds per square inch, or make a fixed gas from any class of cheap oil. Facts seem to point strongly against the success of the internal-combustion turbine working on any of the present cycles, and until a new cycle or a new heat carrier can be found, there is little hope of much progress being made in this direction.

A question that is greatly exercising the minds of engineers is: How big can the Diesel engine be made? Steam installations have already reached the enormous total of 90,000 H.P. concentrated in one ship, and at first sight this would seem quite hopeless to attempt with oil engines. Experiment and the present means of manufacture seem to be the determining factors, and it is probable that the size of the crankshaft and not the size of the cylinder will limit the power of the oil engine. Considering that 2,500 H.P. has been put into one cylinder, I have assumed for the sake of argument that it is possible to put 2,000 H.P. into one single-acting two-cycle cylinder, or 4,000 H.P. into one double-acting two-cycle one. Since the volume of the cylinder rises as the cube of the dimensions and the area of the exhaust ports is roughly proportional to the square of the dimensions, it follows that there is an economic limit to the size of the cylinder on account of the increasing length of the ports in proportion to the stroke of the pistons. This limit is certainly over the powers assumed. By combining twelve single-acting two-cycle cylinders or six double-acting ones, then a total of 24,000 H.P. is reached for one shaft, and if three shafts are used this would amount to 72,000 B.H.P. in one ship. The piston of this motor would be about 45 inches in diameter, and the stroke 60 inches. The crankshaft would be 27 inches in diameter, stressed to 11,000 pounds per square inch. By using high-tensile steel and a judicious arrangement of cranks, it would be possible to increase the power up to 30,000 H.P. for the same diameter of shaft by adding two extra double-acting cylinders, thus giving 90,000 B.H.P. in one ship.

The diameter of shaft given is limited by bearing-pressure considerations; and the power, as already suggested, that can be passed through it, can only be increased by using a better quality of steel, and increasing the number of cylinders in a line, instead of increasing the diameter and stroke of the pistons. The large diameter of shaft in proportion to the throw would necessitate the pins, webs and shaft being forged solid, thereby increasing the difficulties of manufacture. It is certain that the motor warships suggested by Mr. James McKechnie are much nearer realization than most engineers thought possible at the time.

Some of the claims made on behalf of the Diesel engine are not quite accurate. The absence of ignition gear is not an advantage to the engine. but constitutes what is probably its greatest disadvantage. In place of simple electrical ignition gear, there is a very high pressure to deal with in the working cylinder, and a complicated two or three-stage compressor to provide injection air at a pressure of 800 or 900 pounds per square inch. This demands the finest workmanship and the greatest care in handling, and has proved the only weak point in some jobs that have otherwise been faultless.

Because the charge is not admitted until the piston is near the top dead center, it is argued that it is impossible to obtain preignitions. It is only necessary to have a leaky fuel-injection valve in order to obtain them, although in the Diesel engine they are not so violent as in engines working on the Otto cycle, for the reason that the pressures are much higher, and the changes in pressure are not therefore so noticeable. Further, the temperature to cause ignition is only reached when the piston is near the top dead center. The real danger is that the high-pressure air would leak into the cylinder if the needle valve were to stick open, and the piston on rising would enormously increase the compression pressure. At least, to my knowledge, two engines have been completely wrecked by causes which could only be traced to this, and it is, therefore, imperative that Diesel engines of any size for marine purposes should be fitted with relief valves on all the cylinders. Again, there are no violent explosions in the Diesel engine when working, for although the changes in pressure are great they are gradual, and do not give rise to shocks, as is experienced in engines working on the Otto cycle.

One of the greatest advantages of the Diesel engine is that, however small the quantity of combustible matter injected into the cylinder may be, it is certain to burn. Hence no weak charges can be sent into the exhaust system to cause trouble. This fact makes the Diesel cycle eminently suitable for dealing with gaseous fuels, especially for reversing engines, and I think that this is the correct line to work along in developing gas plants for ship propulsion. In a reversing engine working on the Otto cycle there are always one or two cylinders full of mixture, and at the point of reversal explosives are discharged into the exhaust. Then, again, it is wrong to introduce the fuel into the cylinder before the exhaust ports are closed, for no matter how great the precautions or attempts at stratification, it is quite impossible to prevent the escape of fuel during charging. At the finish of charging there is always some gas in the ports and clearances, and this is swept out through the exhaust during the next scavenging period. These things tend to make the efficiency of the large two-cycle gas engine rather poor, and any development must take place by withholding the gas until the exhaust ports are quite closed.

I am not aware if Diesel engines have been run with gas, but all that is necessary to test the idea is a pressure intensifier in conjunction with the existing multi-stage injection air compressor.

CRANKSHAFTS.

Cranks should be arranged with regard to the following points, which are given in order of importance:

1. Uniform twisting-moment.
2. Balance of the engine as a whole.
3. Unbalanced couple between any two or more adjacent cranks to be as small as possible.
4. The angle between two adjacent cranks to be as large as possible, so as not to have them both firing on the down stroke, and causing excessive pressure on the bearing between them. If there are two bearings between each crank this point has little significance.
5. If the shaft is made in pieces, these should be interchangeable, in order to keep down the number of spare parts.
6. In four-cycle work, adjacent cranks should not be arranged on the same center line, on account of concentrating the centrifugal and inertia loads.
7. As far as possible avoid making the maximum twisting and bending moments occur at the same instant.

In crankshaft design it is generally assumed that the maximum moments on the shaft occur when starting and before inertia has had time to come into play, and that this will be due to gas pressures caused by the working

fluid. On account of the manner in which Diesel engines are started, however, this is certainly not the case. In a four-cylinder engine the starting air is admitted over a crank angle greater than 90 degrees, in order that the engine will start from any position, and in a six-cylinder engine the angle will be something more than 60 degrees. The pressure of the starting air is generally about 500 pounds per square inch, and it can be shown that the stress due to admitting air over a crank angle of 90 degrees, as is done in six-cylinder engines, will stress an engine shaft to 14,000 pounds, that would otherwise be only stressed to 8,000 pounds per square inch. This is rather barbarous treatment of shafts, and it would be much better to relieve the compression pressure to 150 pounds per square inch and use starting air through a reducing valve, at that pressure, at the same time effecting a large saving in compressed air. On starting, cards showing pressures of 700 pounds per square inch have been taken from Diesel engines, this high pressure being due to the starting air pressure mounting up before the pistons had time to get under way.

In the following crankshaft formula,

Let d = diameter of shaft, in inches.
D = diameter of cylinder, in inches.
f = stress in pounds per square inch.
P = mean pressure in cylinder, in pounds.
P_m = maximum pressure in cylinder at instant of maximum twisting moment.
= 420 pounds per square inch approximate.
n = number of working cylinders.
$R_m = \frac{\text{maximum twisting moment}}{\text{minimum twisting moment}}$.
$a = \frac{\text{distance between centers of bearings}}{\text{diameter of cylinder}}$.
= 2 for most engines.
$b = \frac{\text{piston stroke}}{\text{diameter of cylinder}}$.
= 1.2 for high-speed engines and
= 1.4 for low-speed engines,

Then
$x = PbnR_m$ for 4-cycle single-acting engines.
$= 2\,PbnR_m$ for 2-cycle single-acting engines.
$= 4\,PbnR_m$ for 2-cycle double acting engines.
$y = \frac{P_m a}{x}$.

Then

$$d = D\sqrt[3]{\frac{x}{f}\left\{.35y + .65\sqrt{y^2 + .4}\right\}},$$

$$\text{or } f = \frac{D^3}{d^3}x\left\{.35y + .65\sqrt{y^2 + .4}\right\}.$$

These formulæ, although rather complicated, take into account all the variables and constants, x and y can be worked out for a particular type of engine, and tabulated for use in design work. It is deduced from the usual equation—

$$M_c = .35\,M_b + .65\sqrt{M_b^2 + M_t^2},$$

the amount of bending allowed for being equal to $\frac{WL}{8}$.

where W = piston load, and L = distance between centers of bearings.

The following notes and suggestions on marine requirements may not be without interest. They refer principally to two-cycle engines.

The use of the stepped-piston engine is not advisable for marine work, chiefly on account of the amount of gear that has to be dismantled before a piston can be examined. In all designs this means the removal of a cylinder, and in some, the cam shaft and gear extending over two cylinders, in addition. The advantages claimed for the design are that balancing is much easier and better, and that if the scavenging part for one cylinder breaks down it only affects the working of that cylinder. The latter reason is quite sound, but with regard to balancing, any addition of weight to the reciprocating mass is bad, and aggravates any unbalanced forces and couples there may be in the engine. In some engines, such as the four-cylinder type with cranks at 90 degrees, it is desirable to have an additional crank, since this can be arranged to give a good balance without disturbing the equal-phase difference of the working cranks. It also makes a cheaper engine to have a separate scavenging crank, although the cylinders do not all receive the same measured quantity of air.

Some of the stepped-piston engines have only one fuel-oil pump for supplying all the cylinders, and if it breaks down or gets choked up, which is by no means uncommon, the whole engine stops working. The better method is to have a separate pump for each cylinder, so designed that it can be overhauled without stopping the engine. Higher powers have been obtained from engines so arranged, because the supply of oil to the cylinders is better under control. This design possesses advantages from the standpoint of governing, which is a most important feature in marine engines. In a six-cylinder engine with one oil pump, one stroke of the pump delivers six charges of oil. Now, supposing that the governor was required to act immediately after the pump had made a delivery, it cannot affect the six charges which have gone beyond its control, and must go through the cylinders, even although the propeller may be out of the water. With six separate oil pumps delivering at intervals to suit the crank arrangement, it would have been able to act upon five of the charges, and would, consequently, be much more sensitive in its action. The oil should be withheld from the Diesel valve until it is just about to open, so that the governor has a chance to act upon it right up to the point at which it is required in the cylinder. With oil engines there is not the slightest fear of racing taking place on account of the propeller lifting out of the water, for it is possible to throw the whole load instantly off the engine, and there is no perceptible variation in speed. The usual method of starting up these engines is to admit air to all the working cylinders, and when a certain speed has been reached, to cut the air off from half the number, and turn the oil on. When this half has commenced firing, the air is shut off from the other cylinders and oil turned on, when the whole engine will be running on oil. Instead of this method all the pumps may be dropped into gear at the same instant that the starting air is turned on, and as soon as firing has commenced the starting air should be shut off all the cylinders.

If the ship is fitted with an auxiliary set of engines, provision should be made for sending the hot water discharged from it through the cylinders of the main engine. This preliminary warming up would prevent the engine being subjected to sudden variations in temperature.

In twin-screw vessels, the gear on each cylinder need not be handed, because each cylinder is a complete unit, and there are no great structural changes such as exist in steam jobs. It is rather a convenience for a man to turn from one engine to the other and find it exactly the same. It also means less first cost, and only half the number of spare parts are necessary.

In order to increase the power for purposes of overload, or in cases of emergency, it is only necessary to create a denser atmosphere in the engine room, by the usual forced-draft appliances, the engines having first been designed to suit the increased compression pressure. With four-

cycle engines this is all that is required, but with two-cycle engines the case is different, on account of the cylinders being open to the outside atmosphere during charging. The back pressure in the exhaust would require to be increased, which can be done by means of a water seal.—"Gas and Oil Power."

OIL-FIRE EXTINGUISHERS FOR NAVAL VESSELS.

By Henry Williams, Naval Constructor, U. S. N., Navy Yard, Brooklyn, N. Y.

The adoption of oil fuel for naval vessels has introduced an element of considerable danger from fire. Due to leakage, which it is practically impossible to prevent, from manholes, pipe connections, pumps and seams in tanks, oil collects in the bilges of the firerooms. The surfaces with which it comes in contact remain coated with a film of oil, which burns freely when ignited or heated, communicating the fire to any oil that may be exposed. This constitutes a danger, and already there have been several fires which, fortunately, have not resulted in serious danger to the ships, though all of them presented that possibility.

The extinction of such fires, once they have gained headway, is difficult. Water has no effect, as the burning oil simply floats on the surface of the water. Sand is more effective, acting to smother the fire. It must cover, however, the entire surface that is burning, and as it does not flow, must be applied locally. This is difficult for portions of the bilges under boilers or other obstructions, and the use of sand thus does not offer an entirely satisfactory guarantee against oil fires.

Recognition of these circumstances led the Bureau of Construction and Repair of the Navy Department, over a year ago, to undertake experiments looking to the development of suitable fire-extinguishing appliances to meet the possibility of such fires and to insure their prompt extinction. A board was appointed to investigate the subject and to undertake the necessary experiments to determine the suitability of the various systems.

The most promising methods that presented themselves to the board were those whereby advantage is taken of a heavy foam mixture which, when applied to burning surfaces of oil, flows over them and smothers the flame by shutting off its air supply. These foam mixtures have been tested extensively abroad, especially in Germany, and have given results uniformly satisfactory. A striking demonstration was made for the board at the plant of the Standard Oil Co., at Bayonne, N. J. A tank about 16 feet long and about 4 feet wide was filled to a depth of 5 inches with naphtha and ignited by an electric spark. The flame, rising to a height of 20 feet, was extinguished in a few minutes by the foam mixture, which formed a blanket over the burning surface.

This foam is produced by the combination of two liquids, which are kept in tanks connected by piping to a mixing chamber. Here the foamy mixture is formed when the liquids are brought together, and sufficient pressure is generated by the chemical action to force the foam out. The volume of foamy mixture generated is about seven or eight times as great as that of the constituent liquids.

Portable extinguishers were designed similar in shape and sizes to the chemical fire extinguishers seen in buildings on shore. These have the liquids in separate compartments and by inverting the extinguisher the foam mixture is obtained and sufficient force generated to throw the foam a distance of about 20 feet. Two extinguishers of this type are supplied in each fireroom of oil-burning vessels. The question is under consideration of making stationary installations of greater capacity in the firerooms of oil-burning torpedo-boat destroyers, and experimental installations of this character are being made.

It is by no means sure, however, that this system of extinguishing oil fires will prove a success on shipboard, though it may be accepted that its efficiency for shore purposes has been demonstrated beyond doubt. One of the most important difficulties that may be encountered is the possible effect on the crew of the carbon-dioxide gas given off by the foamy mixture. Ten per cent. of this gas in air has been known to prove fatal, and continued breathing of 2 per cent. is dangerous. In closed spaces, such as firerooms, where fires are apt to occur, especial care will need to be taken to prevent accumulation of this gas when the apparatus has been used.

Another difficulty, and the one that may render the system ineffective when needed, is the liability to deterioration of the mixtures or the loss by evaporation of the water in which the solutions are made.

The corrosive action of the acid mixture on the containers and piping has been met in shore installations by lead-lining them. The excessive weight required, however, is an obstacle to this means being adopted for shipboard installations, and in those now being fitted up reliance is placed on acid-resisting paint to prevent corrosion. It seems probable that this will not prove adequate and that corrosion of the tanks and piping will take place, and that such valves as are used will be rendered ineffective in a short time.

The formulas for the mixtures are as follows:

LIQUID NO. 1.	Parts by weight.
Glue	1
Glucose	½
Soda bicarbonate	7½
Salicylic acid	¼
Water	100

LIQUID NO. 2.	Parts by weight.
Aluminum sulphate	10
Sulphuric acid	½
Water	100

Carbon-dioxide gas is generated by the mixture of the soda bicarbonate and aluminum sulphate. The glue renders the liquid viscous and causes it to produce the foam. The salicylic acid is added as a preservative for the glucose, which is intended as a stabilizer to prevent the too sudden precipitation of the aluminum by hydroxide. The sulphuric acid aids in starting the reaction and acts also as a stabilizer. It is possible that it may be found that the acid itself can be omitted and in such an event the corrosive effect of the mixture on the container and piping will be reduced considerably.—"Engineering News."

THE QUALITY OF STEAM IN BOILERS.

HOW TO MAKE A CALORIMETER FROM PIECES OF PIPE AND STANDARD FITTINGS; ALSO THE METHOD OF FIGURING THE QUALITY OF THE STEAM AND MAKING CORRESPONDING CORRECTIONS IN THE TOTAL EVAPORATION.

By O. Z. Howard.

An essential feature of a boiler test that is frequently omitted is the determination of moisture in the steam. While this may be so small as to affect the result but little in some cases, in others it may affect the equivalent evaporation as much as 2 or 3 per cent.

If a calorimeter is not available, one can be made very easily from pipe

and standard figures like the illustration. The perforated nipple, with the long thread, should be screwed into a vertical portion of the steam pipe, as near the boiler as possible. The nipple should be of such length that the ⅛-inch holes near the ends will be about ½ inch from the sides of the steam pipe when screwed in place. The calorimeter should be thoroughly lagged with hair felt, from the point where it leaves the steam pipe to the elbow below the lower thermometer well.

The thermometer wells should be filled with mercury or a heavy oil, and a second thermometer should be suspended, in the open air, near each of those in the wells, to obtain the temperature of their stems, for making corrections as shown later. A manometer gage should be mounted on the side of the lower tee for measuring the back pressure in the exhaust pipe. An ordinary U-tube, as commonly used for taking boiler draft, will answer all requirements if filled with mercury in place of water.

If the amount of evaporation is determined by weighing the water before entering the boiler, no account of the loss of steam from the calorimeter will be necessary; but if the steam from the boiler is condensed and weighed, a correction for this loss will have to be made. This can be accomplished by running the exhaust into a barrel partly filled with water, thus condensing the steam and then weighing; by calibrating the orifice for the pressure used during the test, or by calculating the discharge. This latter method is correct enough for all practical purposes, and the discharge in pounds per minute, for an orifice ⅛ inch in diameter, will be $0.0105P$, where P is the absolute boiler pressure.

Before readings are taken on the calorimeter, the ½-inch valve should be opened wide for 10 or 15 minutes, then frequent readings of the four thermometers and the gage should be noted. The reading of the thermometer in the upper well, if properly corrected, should be a check on the gage reading. Assuming that there is no loss of heat by radiation in passing through the instrument, the total heat in the mixture of steam and water before throttling and in the superheated steam after throttling, will be the same, and can be expressed by the equation

$$xr + q = H_c + 0.476\,(T_1 - T_c)$$

or

$$x = \frac{H_c - q + 0.476\,(T_1 - T_c)}{r} \qquad (1)$$

where

x = Quality of the steam;
r = Latent heat of the steam at the pressure in the main;
q = Heat of liquid due to the pressure in the main;
H_c = Total heat in 1 pound of dry steam at the calorimeter pressure (shown by manometer gage);
T_1 = Temperature in calorimeter (shown by lower thermometer);
T_c = Normal temperature of steam in calorimeter due to calorimeter pressure.

THERMOMETER CORRECTIONS.

In general, all thermometers are calibrated for total immersion; that is, for the condition where both the stem and the bulb are at the same temperature. But in taking the temperature of the steam in a calorimeter some of the stem projects into space colder than the bulb. That being the case, a correction must be applied to the observed reading in addition to the correction found by the previous calibration of the thermometer.

The Bureau of Standards gives the following formula:

$$\frac{\text{Stem correction}}{(T - t)} = 0.000092 \times n \qquad (2)$$

where

n = Number of degrees projecting from bath;
T = Temperature of the bath;
t = Mean temperature of the projecting stem.

The mean temperature t may be measured approximately by means of an auxiliary thermometer suspended near by.

If the thermometer is used under conditions such that it has only a small stem correction, the reading of the thermometer corrected by the mean of the table (from previous calibration) may be taken as the temperature of the bath, since a difference in the bath temperature makes a much smaller difference in the value of the stem correction. If, however, the number of graduations projecting from the well and the difference of temperature between the bath and the space above it are large, it will be necessary to determine the stem correction approximately and apply this to the corrected reading of the thermometer in order to get the approximate temperature of the bath. This may be done by using the reading of the thermometer, corrected from the table, as the temperature of the bath, and substituting it in the foregoing formula from which the stem correction can be obtained very closely.

CORRECTION FOR QUALITY OF STEAM.

After determining the percentage of moisture in the steam it is necessary to determine the factor of correction for the quality of the steam, or the factor by which the apparent evaporation, as determined by the boiler test, is to be multiplied in order to obtain the actual evaporation.

Let x be the quality of the steam as determined by the calorimeter; $1-x$ the proportion of moisture in the steam; F the factor of correction for the quality; H, the total heat of 1 pound of steam at boiler pressure; q the heat of the liquid at the temperature due to boiler pressure, and h the heat of the liquid at the feed-water temperature.

The proportional part x receives from the boiler the sensible and the latent heat, or, the total heat in this portion above the temperature of the feed water is $x\ (H-h)$ B.t.u.; and the part $(1-x)$ representing the water in 1 pound of the mixture is $(1-x)\ (q-h)$ B.t.u. But if all the water had been evaporated each pound would have received the total heat in the steam above the temperature of the feed water, or $(H-h)$ B.t.u.

Hence,

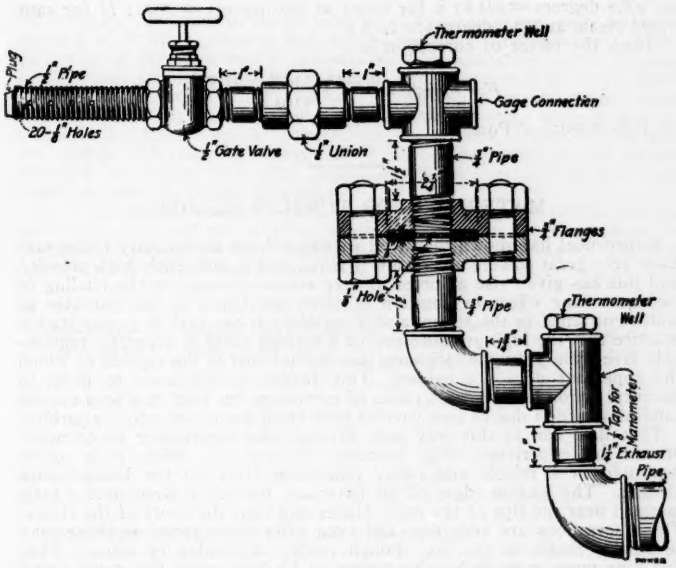
$$F=\frac{x\,(H-h)+(1-x)\,q-h)}{(H-h)}$$

or

$$F=x+\frac{(1-x)\,(q-h)}{H-h} \qquad (3)$$

Example: From a boiler test the following data are obtained with a throttling calorimeter: Steam gage, 144 pounds per square inch; temperature in the upper well, when immersed to the 200-degree mark, 361 degrees F.; auxiliary thermometer at the upper well, 102 degrees F.; temperature in the lower well, 260 degrees, when immersed to 240 degrees; auxiliary thermometer at the lower well, 102 degrees; manometer gage, 0.42 inch of mercury (corresponding to a pressure of 0.21 pound per square inch), and the barometer (corrected) stood at 29.77 inches.

Assume that from the calibration of the thermometers before the test. the reading of the thermometer in the upper well at 361 degrees was 2 degrees too high, and the reading of the thermometer in the lower well at 260 degrees was 1 degree too low. Then from equation (2) the stem



SAMPLING TUBE AND HOMEMADE CALORIMETER.

correction for the upper thermometer is $0.000092 \times 161\ (359 - 102) = 3.8$ *degrees,* and the true reading will be.

$$361 + 3.8 - 2 = 362.8 \text{ degrees } F.$$

As so little of the stem of the lower thermometer projects, the stem correction need not be taken into account, and the true reading will be

$$260 + 1 = 261 \text{ degrees } F.$$

The reading of the barometer, 29.77 inches, corresponds to a pressure of 14.62 pounds per square inch; hence the absolute steam pressure is

$$144 + 14.62 = 158.62 \text{ pounds}$$

per square inch, which, from the steam table, corresponds to a temperature of 362.9 degrees, agreeing within 0.1 degree of the temperature as obtained by the thermometer.

The quality of the steam may now be obtained. From the steam tables: r for saturated steam at 362.8 degrees F. equals 859.6 B.t.u.; q for saturated steam at 362.8 degrees F. equals 334.8 B.t.u.; H_c for 1 pound of dry steam at 14.83 pounds (14.62 + 0.21) pressure absolute equals 1150.5 B.t.u.; T_1 from the corrected temperature in the lower well equals 261 degrees F., and T_c for the steam at 14.83 pounds pressure equals 212.41 degrees F.

Substituting in formula (1),

$$x = [1150.5 - 334.8 + 0.476\ (261 - 212.41)] \div 859.6 = 0.9758$$

or, the steam is 97.58 per cent. dry.

If the temperature of the feed water entering the boiler is 100 degrees in equation (3): $x = 0.9758$ and $(1 - x) = 0.0242$; q for saturated steam

at 362.8 degrees = 334.8; h for water at 100 degrees = 67.97; H for saturated steam at 362.8 degrees = 1194.4.

Then, the factor of correction is

$$F = 0.9758 + \frac{0.0242\ (334.8 - 67.97)}{1194.4}$$

or $F = 0.9812$.—"Power."

MATERIALS FOR TURBINE BLADES.

Experiment has shown that a jet of steam from an ordinary boiler may have very great powers of erosion if moving at a sufficiently high velocity, and this has given rise at times to very serious damage to the blading of turbines. The effect in question is often attributed to the particles of water entrained in the steam, and it certainly is the case in power-station practice that the high-pressure end of a turbine rotor is generally remarkably free from pitting, which only becomes marked in the regions to which the superheat does not extend. This feature would seem to point to chemical action being a main cause of corrosion, but there is also a certain sand-blast action due to lime carried over from the boiler into the turbine.

That lime can in this way pass through the superheater is definitely proved by experience with reaction turbines, in which it is often deposited in a friable and easily removable layer on the low-pressure blading. The leading edges of the latter are, moreover, sometimes a little serrated near the tips of the rotor blades and near the roots of the stator. These serrations are very fine, and even after some years' working may be hardly visible to the eye, though readily detectable by touch. That this lime must exert an erosive action on blading, when the steam speed is too great for it to adhere, is certain. In one case, in which the soft alloys successfully used for reaction blading were adopted for a velocity-compounded wheel, the blades were cut to pieces in a fortnight's run. Here it is possible that entrained particles of lime may have been largely responsible for the result, but in all turbines, even those in which the velocity is as great at the high-pressure end as at the other, the erosion is much more marked in the regions where the turbine is traversed with wet steam—a circumstance which points to corrosion rather than erosion as being the really effective cause of the damage done. No doubt this corrosion is promoted by the rush of the fluid washing off any oxide formed, and thus preventing the formation of a protective layer.

The types of metal best suited for use for turbine blades form the subject of an article contributed by M. Pierre Breuil to a recent issue of "La Technique Moderne." The 63-37 copper-zinc alloy which has been successfully used for reaction turbines is, he states, too soft to be suitable for impulse turbines. One French manufacturer recommends for such uses a 30 per cent. nickel steel, stating that steel containing 25 per cent. nickel becomes brittle, and develops cracks under the influence of the steam. The 30 per cent. nickel steel in question has an elastic limit of 32 to 38 tons per square inch, a tensile strength of 38 to 51 tons per square inch, and an elongation of 40 to 30 per cent. It is, however, costly and difficult to work. The same maker, therefore, recommends, as an alternative, steel with from 3 to 5 per cent. of nickel, which must contain more than 0.4 per cent. of manganese. Such steels, he states, will have a tensile strength of 38 tons with 3 per cent. of nickel, and of 41 tons with 5 per-cent. of nickel.

Another firm suggests the use of electric steels containing little carbon, silica or manganese. These are free from particles of cinder or slag, which serve as centers for the starting of corrosion. As an alternative, the use of a 5 per cent. nickel steel of their make is proposed, which, in

the annealed state, has an elastic limit of 47 tons per square inch, a tensile strength of 48 tons per square inch, and an elongation of 36 per cent. After quenching at 800 degrees C., and subsequently annealing to 600 degrees C., its tensile strength is raised to 66 tons per square inch, its elastic limit to 57 tons per square inch, whilst the elongation is still 9 per cent. Another 5 per cent. nickel steel, furnished by the same makers, has an elastic limit of 25.4 tons per square inch, an ultimate strength of 32 tons per square inch, and an elongation of 23 per cent. As to the results obtained in service with alloy steels, M. Breuil declares he has been unable to obtain reliable information.

As to bronzes, two alloys—one consisting of 72 parts copper and 28 parts pure zinc, and the other of 85 per cent. copper and 15 per cent. of manganese—have been used by Messrs. Melms and Pfenninger. The first has given the following results on test:

Temperature, degrees C.	Elastic limit, tons per square inch.	Ultimate strength, tons per square inch.	Elongation, per cent.
Ordinary	34.9	36.2	11.0
100	34.3	36.2	8.5
200	32.7	34.3	11.0
300	22.2	24.1	30.0
400	7.6	12.7	75.0

The manganese alloy gave the following results:

Temperature, degrees C.	Elastic limit, tons per square inch.	Ultimate strength, tons per square inch.	Elongation, per cent.
Ordinary	38.7	41.3	10.2
100	36.8	40.0	10.5
200	37.5	41.0	11.0
300	33.0	36.8	21.0
400	21.0	24.1	68.0

Both alloys, M. Breuil states, show evidence of having been considerably hardened by cold working; and he remarks that the falling off in strength with temperature would be greatly increased if this temperature were maintained for a long time.

Monel metal is another alloy which has been used for turbine blading. It consists, M. Breuil says, of 70 per cent. nickel, 29 per cent. of copper, and 1 per cent. of iron. Under test the following results were obtained:

.....	Tested at 20 degrees C.		Tested at 300 degrees C.	
Elastic limit, tons per square inch.....	15.3	38.4	12.3	34.0
Tensile strength, tons per square inch	35.0	41.1	19.5	36.7
Elongation, per cent.....	34.5	8.0	23.5	6.0
Contraction of area.....	64.4	60.7	69.0	58.8
Condition	Annealed.	As drawn.	Annealed.	As drawn.

M. Breuil states that here, again, the metal softens considerably at a temperature of 300 degrees C., and that its elastic limit is relatively low, but it has the advantage of being little corrodible.

Metals which owe their hardness to cold-working must, M. Breuil remarks, be considered to be in an unstable condition, and, consequently, the elastic limit in service should not, even in the case of the best, be reckoned as higher than 12¾ tons per square inch. Such cold-worked metals have further, he remarks, a greater tendency to form large crystals under the tempering action of the steam than if used in their natural state. Steels are less liable to be affected in this way; but M. Breuil does not consider it necessary, on this account, to abandon the use of copper alloys, but merely that designers should not count on the maintenance of the artificial strength and hardness produced by cold working. To avoid the risk of gross crystallization being produced by the prolonged tempering action of the hot steam, he suggests that they should be rapidly annealed before use. Cast iron, he suggests, may be used for the guides, but the metal chosen should be of fine texture, and free from graphite particles. A similar metal to that used for the cylinders of large gas engines is, he suggests, suitable. This is very free from phosphorus.

For labyrinth packings M. Breuil states that the lead bronzes are well adapted. These were originated in America for use in the axle boxes of railway cars. The metal contains no less than 30 per cent. of lead. The separation of the latter on cooling is prevented by the addition to the alloy of a little nickel or tin, and by casting it with special precautions. It cannot, however, be forged. Aluminium bronzes are, he states, unsuitable for use in such packings, as they always contain a certain amount of alumina.—"Engineering."

A SUMMARY OF EIGHT YEARS' WORK ON THE PANAMA CANAL.

Eight years ago, on May 4, the United States Government assumed the gigantic engineering undertaking of building a canal across the Isthmus of Panama. This was 23 years after the work had been inaugurated by the French under De Lesseps, and after nearly $400,000,000 had been expended by the French companies.

For the rights of the French company the United States paid $40,000,000. This amount paid for some 78,000,000 cubic yards of excavation and a great variety of equipment, some of which has been in continuous use since, but by far the greater part has been sold as scrap metal. Of the 78,000,000 cubic yards excavated by the French only 30,000,000 cubic yards came within the canal prism as it has ultimately been constructed.

On May 1, 1912, under the direction of American engineers, 171,250,000 cubic yards of material had been excavated and less than 24,250,000 cubic yards remained to be removed, which is being done at the rate of 2,500,000 cubic yards a month. The excavation of Culebra cut is more than 90 per cent. completed. Each year has seen a greater record made in the work of excavation than that of any previous year. From May 1, 1911, to May 1, 1912, the amount of material removed was nearly 16,500,000 cubic yards, and much of this was rock.

The lowest steam shovel cut is now less than 17 feet above the bottom of the channel at Culebra, the summit of excavation. From this point the cut runs to final grade at both ends. At the north end the excavation is practically completed from the Chagres River as far south, toward Culebra, as Las Cascadas; and at the south end the trench is nearly to grade for a mile or more. The greater part of what remains to be excavated lies between Empire and Gold Hill, a distance of about two miles.

On May 1 the great concrete locks at Gatun and Pedro Miguel were

almost ready except for the gates, and the locks at Miraflores were more than 65 per cent. completed, with the work advancing at the rate of 20,000 cubic yards of concrete a week placed.

At Gatun, for the locks and dam nearly 2,000,000 cubic yards of concrete have been made and placed since the work was begun. At the Pedro Miguel locks the total placed is about 840,000 cubic yards, and at Miraflores, nearly 900,000 cubic yards. This makes a grand total of about 3,700,000 cubic yards of concrete, or enough to pave the entire surface of the Island of Manhattan to a depth of 2¼ inches, or to make a single cube 464 feet on a side.

The manufacture of the great steel gates was started in October, 1910, and their erection on the Isthmus began the following June. On March 31, 1912, the erection of these 58,000 tons of structural steel into 46 pairs of lock gates was already one-quarter completed, a task which entails the driving of 6,000,000 field rivets. The contract for machinery for operating the gates was let in June, 1911, and erection was begun in January, 1912; at this writing, one gate at Gatun has already been fully equipped and successfully operated.

Of the rising stem valves, 10 feet 8 inches wide and 18 feet 10 inches high, which control the flow of water into the lock chambers, two had already been put in place on March 31, 43 were ready for erection and the contract for the entire 134 had been let, a total of 1876 tons of metal. The installation of the special cylindrical valves which control the flow of water from the center culverts into the lock chambers is practically completed except for a few in the Miraflores locks. There are 120 valves of this type The contract for the electric motors operating the valves has been let and their erection is under way.

The erection of the first of the six great emergency steel dams which will guard the upper levels of the locks in case of accidents to the gates has been begun at Gatun and the fabrication of the other five is under way in the United States. Each is built essentially of a pair of cantilever trusses, 263 feet long, so pivoted that their longest arm may be swung across the 110-foot locks and the wicket girders which form the dam dropped into place. The total weight of the six dams is 12,000 tons.

The fender chains for preventing unmanageable ships from colliding with the gates and hydraulic apparatus for operating them are in the process of manufacture. The chains are being made at the Boston Navy Yard, and one of them has been completed and recently passed a successful test. The fenders for the approach walls will soon be ready to place in position.

Practically every detail of the lock operation has been worked out. The contract for one of the electric towing locomotives, of which there are to be forty, has been let, and it is now being built in the United States. The electrical apparatus required to operate the locks includes 944 motors, all of 220-volt, 25-cycle, 3-phase, alternating-current type, and of these 803 have already been contracted for. The contract for the hydro-electric plant at Gatun, which will furnish most of the power for the operation of the locks, has been let and work on the station building will soon be started.

The Gatun dam is rapidly nearing completion. The concrete spillway is so far completed as to make possible the closing of the sluice gates, and in the course of a few months Gatun Lake will be at an elevation of 50 feet above sea level. The hydraulic fill on the east section of the great earth dam has been completed and the placing of the dry fill on this section to its final elevation of 107 feet above sea level is progressing rapidly. At the same time the work of paving the south face is well under way. On the west section of the dam three suction dredges are now completing the hydraulic fill, and dry material from borrow pits is being added. The west dam at Miraflores has been practically completed. The east dam there is to be a concrete wing wall and has not yet been

begun. The work on the west dam at Pedro Miguel is also practically completed.

Work on the fortifications to guard the entrances of the canal was begun during the present year. Plans for lighting the canal by beacons and buoys are completed and approved, and work on them has also been begun. Besides all this work in constructing the canal proper, almost the entire line of the Panama R. R. has been relocated and the work of rebuilding on the new location is nearly completed.

Thus at the end of eight years of American industry the end of the great undertaking, without failure in any detail, is actually in sight. Already laborers are being discharged and greatest pressure of work has passed.

To accomplish this the United States had expended up to March 31, 1912, $252,000,000, including the payments to the French company and the Republic of Panama. Excluding this $50,000,000, we have just half the amount expended in 23 years by De Lesseps in a futile attempt to accomplish much less of an undertaking.

The actual amount which has been charged to the department of construction and engineering during the eight years of American work, is $144,600,000. The expenses of the department of civil administration, which includes also the maintenance and operation of public works in the Canal Zone and in the cities of Colon and Panama, the construction and repair buildings and the survey of the Canal Zone amounted to $5,515,000. The department of sanitation, which has contributed in so large a measure to the success of the enterprize, had expended nearly $14,500,000. General items, which include the payments to the French company and to the Republic of Panama, the construction of water works and sewerage in the cities of Colon, Panama and in the Canal Zone, the Panama R. R. expenses, the cost of terminals, lands purchased, etc., amounted to $86,720,000.—"Engineering News."

LIMITING EFFICIENCIES OF STEAM PLANTS.

THE INFLUENCE OF FEED-WATER TEMPERATURE UPON ABSOLUTE EFFICIENCY AND CALCULATIONS SHOWING THAT THE BEST EFFICIENCY ATTAINABLE IS THAT BASED ON THE STEAM CONSUMPTION OF THE MAIN ENGINE WITH FEED WATER AT THE TEMPERATURE OF ITS EXHAUST.

THE LOW THERMAL EXPENDITURE OF OPERATING STEAM PUMPS IS SHOWN WHEN THEIR EXHAUST CAN BE USED TO HEAT THE FEED WATER.

By Prof. R. C. H. Heck.

In every heat-engine plant there are two efficiencies to be considered—one absolute, the other relative. Absolute efficiency is the ratio of work output to heat input. It has two values in every case, first, the actual or realized value; second, the ideal. The latter represents the limit of the attainable (never reached), or the performance of an ideal heat engine under the controlling conditions. Dividing the actual by the ideal performance gives the relative efficiency, sometimes called the efficiency ratio. These general ideas may well be illustrated by an example.

Example 1. Suppose that a condensing plant consumes 12.5 pounds of dry saturated steam per horsepower-hour at 165 pounds absolute pressure corresponding to 366.1 degrees F., from feed water at 149.1 degrees. From the steam tables, the total heat of one pound of dry steam at 165 pounds absolute is found to be 1,195.4 B.t.u., while the heat in one pound

of water at 149.1 degrees is 116.9 B.t.u.—both these quantities being measured above 32 degrees. Then the heat of formation, or that received from the fire per pound of steam is

$$1{,}195.4 - 116.9 = 1{,}078.5 \ B.t.u.$$

As to output, one horsepower for one hour equals.

$$60 \times 33{,}000 = 1{,}980{,}000 \ foot\text{-}pounds$$

of work done. Reducing this to heat units by dividing by 778, the amount of heat converted into work is 2,545 B.t.u. This work being credited to 12.5 pounds of steam, the output per pound is

$$2{,}545 \div 12.5 = 203.6 \ B.t.u.$$

Then the absolute thermodynamic efficiency of the plant is

$$E = \frac{203.6}{1{,}078.5} = 0.1888,$$

or 18.88 per cent.

To obtain the ideal efficiency, the pressure or temperature of the exhaust steam must be known. Let this temperature be 90 degrees, corresponding to a pressure of 0.696 pound absolute, or to a vacuum of 28.50 inches when the barometer is at 29.92 inches. With heat received from the fire after the general manner of the ordinary boiler, or with the Rankine cycle of ideal steam-engine operation, the output would be 334.6 B.t.u. per pound of steam. (This quantity is the difference between the total heat in one pound of the steam at 165 pounds absolute and the heat after adiabatic expansion to 0.696 pound absolute.) If the ideal output, or the best that could be done under perfect conditions of working, is 334.6 B.t.u. per pound of steam, while the actual output is 203.6 B.t.u., the relative efficiency will be

$$E_r = \frac{203.6}{334.6} = 0.608,$$

or 60.8 per cent.

It will be noted that only that part of the plant is considered which has to do with the use and application of heat, after it has been received from the fire. Boiler and furnace efficiency, in generating heat by combustion and in transferring it to the working medium (water and steam), is another matter.

LIMIT OF ABSOLUTE EFFICIENCY.

The present purpose is not to show how ideal performance and efficiency are derived by logical processes, but to keep much nearer the practical side of the problem and consider how actual efficiency is related to feed-water temperature, and how the latter is determined by the combined influences of the main engine and of auxiliaries. Assuming the steam consumption to be fixed by operating conditions which are outside of this discussion, the following proposition is to be established:

If the heat of steam formation be determined from feed water of the temperature of the main exhaust, and if the steam consumption of the main engine alone be used in calculating the output, the resulting thermodynamic efficiency is the highest of which the plant is capable.

The best that can be done with any combination of auxiliaries and feed-water heater is to approach this upper limit. As stated, the proposition meets the simpler case in which the main engine can return heat to the boiler only from its exhaust; a broader specification, covering such action as the return of hot water from steam jackets, will be dealt with later.

PLANT WITH STEAM-DRIVEN PUMPS.

Example 2. Continuing the controlling data of the first example, let the 12.5 pounds of steam per horsepower-hour be divided into 11.8 pounds consumed by the main engine and 0.7 pound by the feed and condenser pumps. Then the limiting efficiency as just defined would be calculated for a steam consumption of 11.8 pounds, from water at 90 degrees. The heat input per pound of steam is the total heat minus the heat in the feed water, or

$$1195.4 - 57.9 = 1137.5 \ B.t.u.$$

while the output is

$$2545 \div 11.8 = 215.7 \ B.t.u.$$

then the best efficiency is

$$E_{max} = \frac{215.7}{1,137.5} = 0.1896,$$

or 18.96 per cent.

Suppose, first, that the exhaust from the pumps is turned into the main condenser or otherwise wasted, and that water is fed to the boilers at a hotwell temperature of 90 degrees. Under this condition the whole 12.5 pounds of steam must be charged, cutting the output to 203.6 B.t.u., while the input remains 1,137.5 B.t.u. The resulting efficiency then is only

$$E = \frac{203.6}{1,137.5} = 0.1790,$$

or 17.90 per cent.

Of course, no properly arranged plant would operate in such a fashion; instead, the exhaust from the auxiliaries would be used to heat the feed water. Suppose that these pumps consume 45 pounds of steam per horsepower-hour of their own development, or convert into work

$$2,545 \div 45 = 56.5 \ B.t.u.$$

per pound of steam. Assume, further, that 25 B.t.u. are lost by radiation from each pound of steam passing through the pumps. Then the total heat carried by the pound of pump exhaust will be the original heat, 1,195.4 B.t.u., less 56.5 turned into work and 25 lost, or

$$1,195.4 - 81.5 = 1,114 \ B.t.u.$$

Temporarily disregarding the need of removing cylinder oil, let 0.7 pound of this exhaust combine with 11.8 pounds of hotwell water at 90 degrees; the resulting water will have some temperature t, to be found by equating the heat from the steam with the heat absorbed by the water, or by the equation

$$0.7 \ [1,114 - (t - 32) \ ] = 11.8 \ (t - 90),$$

whence

$$12.5 \ t = 1,864, \text{ or } t = 149.1 \ degrees.$$

This is the feed temperature which was used in the first example.

An instructive comparison can be made by considering the heat input per horsepower-hour under different conditions. Thus for the main engine alone, which uses 11.8 pounds of steam and is capable of returning feed water at 90 degrees, or which requires 1,137.5 B.t.u. per pound of steam, the heat received is

$$1,137.5 \times 11.8 = 13,420 \ B.t.u.$$

For the combination of engine, pumps and feed-water heater, which uses 12.5 pounds of steam from water at 149.1 degrees, the corresponding heat supply is

$$1,078.5 + 12.5 = 13,480 \ B.t.u.$$

From these numbers can be obtained the same efficiencies that have already been worked out with heat quantities per pound of steam, namely:

$$E_{max} = \frac{2,545}{13,420} = 0.1896,$$

and

$$E = \frac{2,545}{13,480} = 0.1888,$$

as in the first example.

The difference of 60 B.t.u. between the numbers 13,480 and 13,420 represents the thermal cost of operating the auxiliaries. It is about 0.7 of the 81.5 B.t.u. taken as the combined useful conversion and radiation loss of the steam. Comparing it with the supply 13,420, the percentage cost of the auxiliaries is only

$$60 \div 13,480 = 0.0045,$$

or 0.45 per cent., as against an apparent steam cost of

$$0.7 \div 12.5 = 0.056,$$

or 5.6 per cent.

In calculating these results, several minor sources of loss have not been taken into account. First of all, water from the condenser at full exhaust temperature—whether condensate or new water passed through a surface heater ahead of the condenser—is hardly attainable; there will probably be a deficiency of at least five degrees, and it may be much larger. An oil separator will remove from the pump exhaust considerable water, which will carry some heat with it. Radiation from the feed pump and feed piping will also cause a small loss.

POWER-DRIVEN PUMPS.

The scheme which has been under discussion may now be compared with that of using power-driven pumps. These are either coupled to the main engine, as in many pumping-engine and marine plants, or are driven by motors as in some steam-electric plants. Whatever power they absorb is either directly or indirectly subtracted from the useful output of the unit. So far as the engine side of the plant is concerned, elimination of steam-driven pumps means the return of feed water at or below exhaust temperature. In the second example the first case, that of 11.8 pounds of steam per horsepower-hour, would correspond to the feed water at a maximum temperature of 90 degrees; but, as just stated, the efficiency of 0.1896 or a little less at the engine would be diminished to the extent of the power consumed by the pumps. The use of a live-steam heater or of an injector, either one giving the boiler the advantage of hot-feed water, would not appreciably affect the thermodynamic relations, since both simply take heat from the boiler and put it back again. If an economizer be included in the plant, it would be considered a part of the boiler, sharing in the function of abstracting heat from the fire.

The contrasting points of the two types of arrangement may be summarized as follows:

With steam pumps, provided that they are not excessively wasteful, all their exhaust may be condensed by the main feed, and the heat rejected by the pumps may thus be returned into the main circuit. Consequently, the power developed and applied by these auxiliaries costs only its own thermal equivalent, plus secondary losses in oil removal and in radiation. If the latter could be reduced to zero, these minor members of the plant would have unit efficiency.

With power pumps, the percentage of power consumed will be much less than the percentage of steam used by the other type of auxiliaries; but since this power is produced at only the efficiency of the main engine,

say from 17 to 22 per cent. in good practice, its thermal cost is much higher than in the other case.

The conditions assumed in the example correspond with a pump power equal to about 1.6 per cent. of that of the engine, as may be shown by the following example:

With independent steam-driven pumps, the pump output per engine horsepower requires 0.7 pound of steam, at the hourly rate of 45 pounds per pump horsepower. Using the power developed at the engine rate of about 12 pounds per hour, the work of the pumps would be done by

$$\frac{12}{45} \times 0.7 = 0.19 \textit{ pound}$$

of steam, which is 1.6 per cent. of 12 pounds.

One and six-tenths per cent. of 2,545 B.t.u. is 40 B.t.u.; and with a thermodynamic efficiency of 0.19 for the engine, this would require a heat supply of more than 200 B.t.u. This is high compared with the 60 B.t.u. deduced in the second example for steam-driven pumps with complete utilization of the pump exhaust.

FEED TEMPERATURE.

In short, what has been discussed is the influence of feed-water temperature upon thermodynamic efficiency. The point emphasized is the determinative influence of the main engine, or of the temperature which it alone (with its condenser) can give to the water delivered to the boiler. In the simpler cases the limiting or ideal feed temperature from the engine (never quite attained) is that of the exhaust steam. If the engine has steam jackets or reheaters in its receivers, and can therefore return hot jacket water, the ideal feed temperature is higher.

Example 3. With the engine using 11.8 pounds of steam per horsepower-hour, let 10 per cent. be condensed in the jackets and reheater, for simplicity at the full initial temperature of 366.1 degrees, the exhaust temperature remaining 90 degrees. What will be the ideal feed-water temperature due to the engine?

Solution. At 366.1 degrees the heat in one pound of water is 338.1 B.t.u. above 32 degrees, or

$$338.1 - 57.9 = 280.2 \textit{ B.t.u.}$$

above water at 90 degrees. The 1.18 pounds of jacket water, if there were no lowering of its temperature by radiation, would carry 330.6 B.t.u. above 90 degrees. This would raise the whole 11.8 pounds of feed water (including that from the jackets) through

$$330.6 \div 11.8 = 28.1 \textit{ degrees,}$$

or to 118.1 degrees Fahrenheit, which is therefore the ideal feed temperature sought. Diminishing the input by 28.1 B.t.u., or from 1,137.5 to 1,109.4 per pound of steam, while keeping the output at 215.7, the limiting efficiency is raised to

$$E_{max} = \frac{215.7}{1{,}109.4} = 0.1945.$$

—"Power."

ELECTRICAL ENGINEERING.

There is a little town of 2,000 people away back in the bush of New South Wales, which can boast of an antiquated electrical light station, under the charge of a "chief electrical engineer," appointed by the aldermen of the district, and simultaneously performing the duties of stoker, wireman and engine driver. He is also the leading and only electrical contractor in the town, and for some years past has reaped a very hand-

some income by selling metal filament lamps to the farmers and other consumers at about 6s. each. It happened not long ago that one of the farmers came to town, and was very much struck at the ridiculously low price for which he could obtain metal filament lamps in the city, and on his advice the enterprizing concern from which he obtained his lamps sent a traveler up to his town, and, in the absence of the previously mentioned "chief electrical engineer," disposed of some few hundred lamps at the ordinary selling price. When our friend returned to his official duties he realized that the sale of his expensive lamps had been considerably handicapped. But he was previously a marine engineer, and we all know that marine engineers are not to be easily cornered. The same evening, therefore, he thought it advisable to boost up his voltage, until every lamp in the town popped off. The next morning he took a push cart around town heavily laden with his own lamps, and told the poor farmers that if they would speculate on these inferior cheap German lamps they would have to stand the consequences.

CHART FOR CALCULATING BOILER TESTS.

By F. O. Pahmeyer.

The accompanying chart has been constructed to facilitate working up results from a boiler test, and the closeness with which results can be determined will be within the limits of the error of the observations making up the test records. By use of the chart, the horsepower, actual evaporation per pound of fuel, equivalent evaporation and efficiency may be determined direct from the data of a boiler test, without going into laborious calculations. Examples are given illustrating the use of the chart, and several of the processes of which determinations are made are as follows:

CAPACITY.

Knowing the average steam pressure and the average feed-water temperature, the factor of evaporation can be ascertained from a table or by calculation. The only other item that must be known is the amount of water evaporated per hour.

From the point on the lower scale, representing the amount of water, follow upward the slightly inclined line to its intersection with the horizontal line representing the factor of evaporation found on the left-hand scale. Follow vertically through this point of intersection to the top scale, the reading of which will be the required capacity in boiler horsepower. One horsepower is defined as the evaporation of 34½ pounds of water from and at 212 degrees F. per hour.

By following down instead of up from the point of intersection, the pounds of water from and at 212 degrees per hour may be read off on the bottom line.

Example.—Assume that 10,250 pounds of water per hour have actually been evaporated, that the average temperature of the feed water is 110 degrees, and that the average steam pressure by gage is 140 pounds per square inch. The factor of evaporation for this temperature and pressure is 1.15. The point of intersection, as just described, carried to the top scale, reads practically 341 horsepower. This point of intersection, carried down to the bottom scale, reads practically 11,800 pounds, which is the equivalent evaporation of water per hour from and at 212 degrees F.

ACTUAL EVAPORATION PER POUND OF FUEL.

From the point on the lower scale representing the amount of water, follow vertically to the intersection with the horizontal line representing the amount of coal. This intersection will fall on one of, or more likely

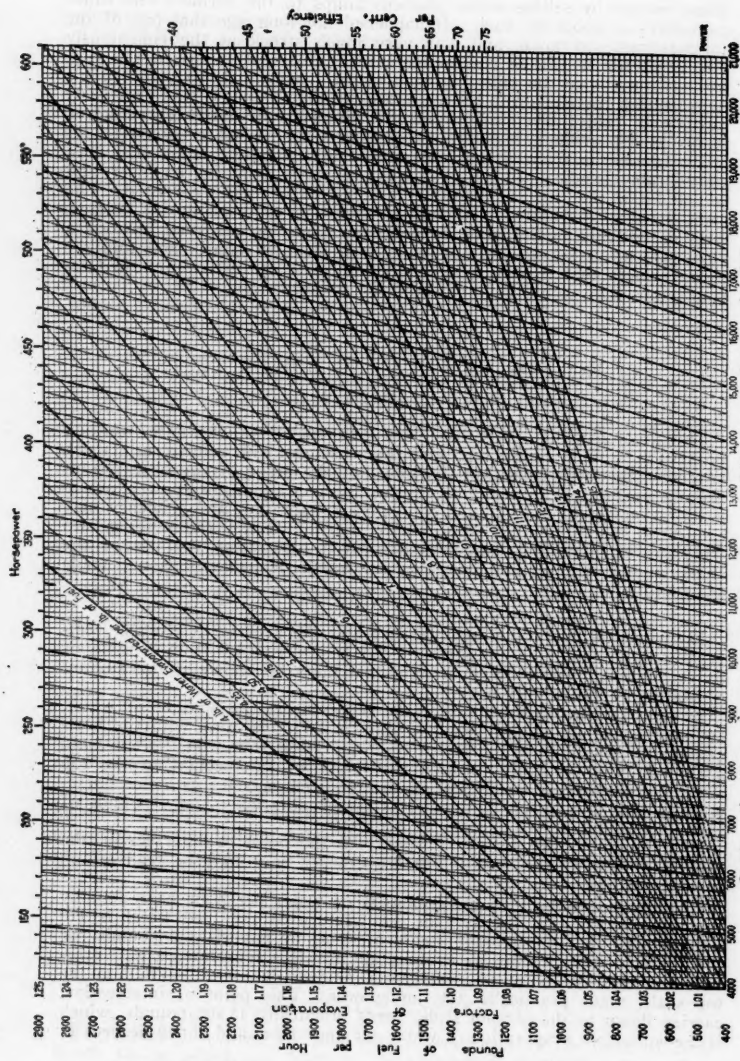

Per Cent. Efficiency
Horsepower
Pounds of Water Evaporated per Hour, from and at 212° F.
B. T. U. per Pound of Fuel
Pounds of Water Evaporated per Hour
Pounds of Fuel per Hour
Factors of Evaporation

CHART FOR CALCULATING RESULTS FROM A BOILER TEST.

between two of, the inclined lines representing the evaporation per pound of fuel. If between the lines, the reading may be interpolated.

Example.—Assume that 10,250 pounds of water have been actually evaporated and 1,580 pounds of coal have been fired; it is immaterial whether this is for one or more hours. The point of intersection, as previously found, falls practically on the line representing 6.5, which is the actual evaporation of water per pound of coal as fired.

EQUIVALENT EVAPORATION FROM AND AT 212 DEGREES PER POUND OF FUEL.

The actual amount of water evaporated, the total amount of fuel burned, the average feed-water temperature and the average gage pressure of the steam for any period must be known. Determine the factor of evaporation and find the equivalent evaporation from and at 212 degrees as described under "capacity." Follow vertically or down from this point of intersection to the horizontal line through the amount of fuel consumed, which point of intersection will fall on one of, or between two of, the much inclined lines, which represent the water evaporated per pound of fuel.

Example.—Assume that 10,250 pounds of water have actually been evaporated and that 1,580 pounds of coal have been burned in the same period of time. Let the temperature of the feed water be 110 degrees, and the steam pressure 140 pounds per square inch; these conditions corresponding to a factor of evaporation of 1.15. The equivalent evaporation from and at 212 degrees, as found under "capacity," gives 11,800 pounds, although it is not necessary to follow down to the bottom line. The intersection, with the amount of coal burned, falls between the lines representing 7.25 and 7.5, which would be estimated at 7.45, the amount of water evaporated from and at 212 degrees per pound of coal as fired. This same method may be followed to determine the evaporation per pound of dry coal if the moisture is determined and deducted from the total weight of coal, and as well for the evaporation per pound of combustible if the moisture and ash are deducted from the weight of coal as fired, and the remainder used as the amount of fuel burned.

EFFICIENCY.

From the point on the bottom scale representing B.t.u. per pound of fuel, follow vertically to the much inclined line representing the equivalent evaporation from and at 212 degrees per pound of fuel. From this point of intersection follow horizontally to the efficiency scale at the right-hand side.

Example.—Assume that the coal burned has a heat value of 12,100 B.t.u. per pound and that the equivalent evaporation from and at 212 degrees is 7.45 pounds of water, in accordance with the preceding examples. The point of intersection will be on a line, which will follow horizontally so as to give a reading of 60 per cent. on the efficiency scale.

REMARKS.

In the foregoing examples it is assumed that there is no moisture in the steam. If it is desired to take any possible entrainment into consideration that quantity must be determined and the total amount of water evaporated decreased by the amount of moisture so found, or correction can be made by reducing the factor of evaporation by the same percentage. For example, if the total amount of water evaporated is 10,250 pounds and the moisture in the steam is 2 per cent., the true amount of water evaporated would be

$$10{,}250 \times 0.98 = 10{,}045 \textit{ pounds},$$

which quantity should be used in place of 10,250 in the determinations. Or the factor of evaporation could be reduced by 2 per cent., making it 1.127, which should be used instead of 1.15. The final result would be the same in either case.

Where the quantities of water and fuel determined are beyond the limits of the diagram, results may be determined by dividing these amounts by two, three or any other factor that may be necessary, then proceed as before. To determine the capacity the result found must be multiplied by the same factor that was used in dividing. The results for evaporation per pound of fuel must not be again multiplied, presuming, of course, that the fuel is reduced in the same proportion as the water.

The chart will give results sufficiently close for all but very refined determinations, and will be found of great assistance in quickly arriving at results during a test, and for the use of those who have not the time or detailed knowledge to perform the various calculations involved.—"Power."

THE SPONTANEOUS COMBUSTION OF COAL.

The chief engineer surveyor of Lloyd's has just issued a useful memorandum on the spontaneous combustion of coal, particularly as it affects the bunkers and holds of ships.

The first report dealt with is that of the Royal Commission of 1875. As a whole, in the light of later knowledge, it may be said that the Commissioners were more wrong than right in their conclusions. They were guided largely by witnesses, who had preconceptions as to the causes of spontaneous ignition, and, although Dr. John Percy and Professor (later Sir Frederick) Abel were perfectly correct in their observations on the absorption of oxygen by coal accompanied by the development of heat, they were wrong about the influence of pyrites—possibly the weight of evidence influenced even their high scientific judgment—and were mistaken in believing that some coals, by their pyrites contents, were exceptionally liable to heating. As they attributed much of the heating to pyrites, for the rapid oxidation of which water is needed, they concluded that damping coal was a practice to be deprecated, but they did recognize that, on the other hand, water would arrest the heating due to the spontaneous oxidation of the carbon itself.

A few years after the publication of the report of this Royal Commission M. Henri Fayol, engineer director of the Collieries of Commentry and Montvicq, wrote a long account of a number of striking observations and experiments he had made on the spontaneous combustion of coal. He confirmed certain chemical principles which had been formulated by Richter—and actually placed before the Royal Commission—and laid the basis for the established opinions amongst scientific people of the present day upon the subject. M. Fayol worked with eighteen kinds of coal, and he found that they all behaved in much the same way. He showed that under the action of air at all temperatures between 25 degrees and 400 degrees C. they at first lose part of their weight, then more than recover it, due to the absorption of oxygen, and then at the end of a certain time at a high temperature a progressive diminution of weight sets in. The power of coal to absorb great quanties of oxygen—as much as a hundred times its own volume—was brought out by these tests.

An investigation of the spontaneous ignition of coals at Commentry was then entered upon, the most striking part of which was made with quite small quantities. These were made up into conical heaps containing about three cubic yards. They were pre-heated to about 100 degrees C. and then covered with a bell. At the bottom and the top of the bell there were holes by means of which the admission of air to the coal within could be regulated. If all the holes were closed the temperature of the parcel of coal fell; if some holes were opened, it rose continuously till spontaneous combustion occurred. By alternately open-

ing and closing the holes the temperature could be made to rise and fall. Thus it was shown that air was essential to spontaneous combustion, but the air had to be in limited quantities; if the bell were lifted the coal, as might be expected, gradually cooled down to the atmospheric temperature. M. Fayol also tested coals in all degrees of wetness, and came to the conclusion that moisture was rather advantageous than otherwise, and, as for pyrites, he was unable to find that it had much effect on spontaneous combustion, and he thought its influence of small importance. It is the absorption of oxygen from the air that is really serious, and, as that is a thousand times greater with small coal than large coal, small coal is more subject to heating than large pieces. The conditions which tend to prevent heating are therefore large size of coal, low temperature, a small volume, and either the complete absence of air or very thorough ventilation. The objection to large masses of coal is, of course, the difficulty of adequate ventilation.

Of course, M. Fayol's observations were all made with special regard to the storage of coal ashore. The question presented by the carrying of coal in bunkers and holds of ships is very different, for, as the author of this memorandum points out, complete ventilation is wholly impracticable, and it is almost as difficult to see how the absolute exclusion of air is to be effected. In fact, the chief engineer surveyor has no suggestions to make as to the safer storage of coal in holds. He, however, summarizes the knowledge of the subject as it applied to bunkers. He corrects the mistaken impression that spontaneous combustion is brought about by the explosion of the gases given off by the coal, enforces the fact that it is the absorption of oxygen that causes the trouble, and that this absorption takes place the higher the temperature; advises that the lower doors of bunkers should be kept shut as much as possible, that the coal likely to be heated by proximity to boilers, uptakes, &c., should be worked out early on the voyage; impresses the fact that danger of overheating increases with the length of time the coal remains in the ship, and advises the damping of small coal of a known fiery nature when it is being charged into the bunkers. Finally, he adds, "Although some kinds of coal have proved to be more liable to spontaneous heating and combustion than others, it should be realized that all coal, even anthracite, is liable to these occurrences."—"The Engineer."

SHIP-PROPELLING MACHINERY OF ALTERNATIVE DESIGNS.

The report of the discussion of the relative merits and cost of different types of ship-propelling machinery which has taken place at successive meetings of the North-East Coast Institution of Engineers and Shipbuilders has just been published, and the amount of comparative detail which was given in the course of the debate thoroughly justifies, because of its suggestiveness, the contention we have put forth on many occasions in favor of the comprehensive consideration of papers which, starting from a common basis, set out the claims of alternative designs. The three papers read before the Newcastle Institution have already been published in "Engineering" (vol. xciii, page 709), and were admirably conceived. Mr. E. L. Orde dealt with the Diesel oil engine in comparison with the triple-expansion steam engine. Sir Charles Parsons and Mr. R. J. Walker similarly compared the geared turbine with the steam reciprocating engine, while Mr. A. C. Holzapfel dealt in a corresponding way with the suction-gas engine. The aim was to consider the comparative suitability of various systems for a typical cargo ship. Thus the range of discussion was very properly limited, and opinion was more likely to be crystallized. As Dr. Thearle has said, "the great bulk of our larger cargo steamers lie between 3,000 and 4,500 tons and measure between 300 feet and 400 feet

long, with a speed of from 8 to 11 knots," and it was well, therefore, to take a ship within such limits. There was a slight difference in the displacement of the ships considered in the respective papers; but this was modified in the course of the debate. The particulars given were for a vessel of 4,655 tons gross, and 2,930 tons net, register, having a radius of action of 3,500 nautical miles, and a capacity for taking 7,880 tons of freight-paying cargo. The average sea speed assumed was 10½ knots, which required that the triple-expansion engines should develop 2,400 indicated horsepower, the geared turbines 2,150 shaft horsepower, and the Diesel engine 2,150 brake horsepower. The dimensions given are those of the accepted typical ship with triple-expansion engines. The aim of the authors and of those who took part in the discussion was to arrive at the differences in the qualities constituting earning capacity, and in the capital and working cost, due to the influence of the type of propelling machinery adopted.

As regards capital value the steamship, a single-screw ship, cost £63,000, provided no superheaters were added to the triple-expansion engines, or £65,000, if this economizing element were introduced. The Diesel-engined ship, with twin engines and propellers, was estimated to cost £78,000, and the ship with turbines using superheated steam, and working through gearing on the propeller shaft, £66,000, and without superheater, £2.000 less. There were sufficient data to enable these figures to be arrived at as a direct result of experience, but in the case of the ship driven by suction gas the step from the small ship, now running, to the large cargo vessel, was reckoned somewhat too large to justify the same close estimate of results. Economy in fuel, of course, reduces the amount of storage needed, and increases the cargo capacity, and therefore reduces the capital cost per ton of freight-earning cargo. The difference of the fuel weight and of bunker capacity for a voyage is considerable. It is estimated that the triple-expansion engine using superheated steam would require 480 tons of fuel for a 3,500-mile voyage; the geared turbine, using superheated steam, 375 tons of coal; and the Diesel engine from 157 to 176 tons of fuel. The difference in weight of fuel and in its stowage thus enables the cargo carried to be higher alike in the geared-turbine ship and in the Diesel-engined ship, so that the capital cost per ton of freight-earning cargo works out at £8.13 for the reciprocating-engined ship using superheated steam, £9.1 for the Diesel-engined ship, and £8.05 for the geared-turbine ship using superheated steam.

The fuel consumption proved a difficult subject on which to find definite agreement. There is first the difference in the system of measurement of power developed. Secondly, while with triple-expansion engines there is a great accumulation of data available, little that is definite regarding oil-engine sea performance has yet been collated. Mr. Orde took 1.6 pounds per indicated horsepower as the average coal consumption for all purposes in his steamer with triple-expansion engines using saturated steam, and certainly this cannot be said to err on the side of economy, as there are many ships which, even without superheaters, are working at 1.3 pounds and 1.4 pounds per indicated horsepower for all purposes. The estimate of 0.55 pound of oil per brake horsepower per hour for the Diesel engine was questioned by many of the speakers. For a two-stroke cycle Nuremberg engine the figure given was 0.44 pound per brake horsepower; for the four-stroke cycle engine of the *Vulcanus*, 0.42 pound; for the two-stroke cycle engine of the *Toiler*, 0.44 pound; and for the four-stroke cycle engine of the *Selandia*, 0.473 pound on the assumption of a thermodynamic efficiency of 76.8 per cent. There is, however, the question as to whether these various results include the fuel for all purposes, for working, for instance, donkey boilers, for galley purposes, etc. It is still matter for doubt whether, as Sir Marcus Samuel seems to claim, the oil consumption of the Diesel-engine ship is one-fourth per unit of weight

of the coal consumption of the triple-expansion steam engine, or one-third, as many contend.

The significant fact was mentioned that there are 600 ships at sea using superheated steam, and that the economy from these ranges from 15 per cent. to as much as 25 per cent., according to the degree of superheat. The most recent proof of the advantage of superheated steam, although it is from torpedo-boat destroyers, is sufficiently striking to justify inclusion here. In the British destroyer *Attack*, fitted with Yarrow's system of superheating, the shaft horsepower developed on the eight hours' full-speed trial was 17,920, and the oil consumed 64.9 tons in the eight hours; while in the case of the *Firedrake*, without superheater, the shaft horsepower developed was 19,174, and the oil consumed 76.9 tons in the eight hours. This shows an economy due to superheating at full speed of about 9.7 per cent. At the cruising speed, the power for 13 knots speed in the *Attack* was 877 shaft horsepower, and the *Firedrake* 779 shaft horsepower, the oil consumed during the 24 hours' trial being respectively 17.4 tons in the case of the *Attack* and 22.1 tons in the case of the *Firedrake*, showing an economy at the slow speed of no less than 33 per cent. due to superheating. In a geared-turbine ship the economy due to superheating, as compared with triple-expansion engines using saturated steam, was stated at 4.9 pounds of water per hour, equal to 32 per cent. Even with the triple-expansion engine the best thermo-dynamical efficiency has not yet been got; while with the high-speed turbines, working through gearing, there is still room for further improvement, particularly should commercial conditions enable oil to be used as fuel for the steam boilers.

The advocates of electric transmission had much to say against helical gearing, and Mr. W. L. R. Emmett, New York, referred to the stress on toothed surfaces increasing with the diminution of the number of teeth in the pinion, making it difficult to use small pinions, and consequently difficult to get large ratios of speed reduction; to the limitation of the length of pinions, and, consequently, the output for a given diameter, owing to the torsional elasticity of the metal, which might prevent equal distribution of load throughout the entire length; to the augmentation of shock and strain on toothed surfaces by increase of peripheral velocity, thus limiting the turbine speeds for a given pinion diameter and output; and to increased size and diameter of gears in ships which were subject to working in a seaway and to vibrations, necessarily conducing to irregularity of strain on toothed surfaces. But all of these mechanical difficulties were, Sir Charles Parsons stated, imaginary difficulties and objections which had not been met with in the design or the working of gears. It was further stated that from experiments which had been carried out in order to arrive at the ultimate strength of the teeth of helical gear it was found that, with pressures as adopted in the *Vespasian*, the ultimate breaking strength was 80 times the load. As regards the maximum power which may be transmitted through gearing, Sir Charles did not seem to anticipate any difficulty in the case of any powers within the limits likely to be reached for many years, and found justifiable satisfaction in the fact that 100,000 horsepower of geared turbines were in use or building at the present time for merchant and naval ships. Indeed, one of the outstanding features of the discussion was the favor with which the Parsons geared turbine was accepted.

If the consumption of the Diesel engine be taken at 0.49 pound per brake horsepower per hour, it is found that the fuel consumption per day for the typical cargo-ship design is 11.3 tons, for the geared turbine using superheated steam 27.2 tons, for the triple-expansion engine using superheated steam 34.5 tons, and for the triple-expansion without superheated steam 42 tons. In no case, therefore, is the fuel consumption of the Diesel engine as low as one-fourth that of the steam engine using coal. The commercial question, therefore, is first, whether the relative cost of coal and of oil on the routes traversed by the ships gives an economy in

fuel charges. This, indeed, is the crux of the question; because, as we have before pointed out, the mechanical problems can be solved when ship-owners are convinced of the superiority of one or other type of ship propulsion. Sir Marcus Samuel stated that tanks and supplies already existed for the whole of the Eastern hemisphere, and that large supplies were kept in hand at Port Said, Suez, Colombo, Bombay, Madras, Calcutta, Rangoon, Penang, Singapore, Bangkok, Batavia, Sourabaya, Hong Kong, Shanghai, Yokohama, Kobe, Adelaide, Melbourne and Sydney. He, moreover, gave the undertaking of his company to increase these as occasion should arise, and stated further that the storage and supplies running at the present moment gave 100,000 tons of oil available at the producing ports of Balik Pappan and the Island of Tarakan, while for the trade west of the Suez Canal tankage existed at Constanza and Batoum. Sir Marcus sought to prove that the present price of oil was abnormal owing to the insufficiency of oil-carrying ships, to the delay in the completion of such vessels now in hand, and to the difficulty of placing orders at reasonable rates for new ships. Thus freights were inflated and the price of oil unduly affected. This he felt sure would be rectified at an early date, with a consequent reduction in the price of oil. He stated that "his company was prepared to make contracts for oil at all the ports where their fields were favorably placed geographically at prices which would leave shipowners a profit if the ratio of four to one (coal consumption to oil consumption) were accepted, leaving the enterprising members of that class the enormous advantages of the oil-driven ship."

The question of increasing the supply of oil was also dealt with, and the oft-repeated assumption that crude oils were suitable was controverted. Mr. E. Hall-Brown thought a considerable amount of assistance in providing a supply of a suitable oil would probably be obtained if gas-works managers of this country and others could see it to be to their advantage to make smokeless fuel for domestic purposes, and to use the tar and tar oils for the production of liquid fuel, the gas generated being available for power and heating purposes. There was considerable difficulty, as Mr. William P. Durtnall pointed out, in constructing an internal-combustion engine that would burn crude coal or residue coal in the form of tar oil. But the addition of 10 per cent. of mineral oil to form the flux for burning the tar oil had proved feasible in the case which he had tried. Crude oil was not only expensive, but unsatisfactory. Texas crude oil has an asphalt basis, and it has been found that after burning it in the cylinder it sometimes leaves a residue which clogs the piston rings and fuses them solid with the pistons. The Borneo oil has a carbon base; but even this oil requires to be treated before use in the Diesel engine. Sir Marcus Samuel stated that where pistons did foul the introduction of half a pint of kerosene into the internal-combustion engine immediately scavenged the refuse. Nevertheless, he contended that it was important that the pistons should be so constructed as to be easily examined and the parts cleaned, because there was risk with some oils of fouling the pistons. Oil engines should be so designed that one cylinder could be disconnected for cleansing at any time. The process of distillation, which gave a distillate oil suitable for so-called crude-oil engines, was undertaken to get valuable products, the oil suitable for engines being practically a by-product, and, as Sir Boverton Redwood has pointed out, only a small proportion of the crude oil is available for use as a source of power. Even if the whole of the crude petroleum, on the basis of the total output in 1910, were utilizable, it would not—allowing for its higher thermal efficiency—replace much more than about 5 per cent. of the world's output of coal, whilst if used in internal-combustion engines it would be equivalent as a source of power to about 15 per cent. of the coal.

There was a wide difference of opinion as to the probable cost of wear and tear. This was assumed by Mr. Orde to be £110 per month for the

oil engine in the type ship dealt with, as compared with £100 in the steam engine. Some of the advocates of the oil engine, however, consider that this was too high, and that the oil engine would have the advantage because boilers required scaling and cleaning, and frequently made a big charge against the steamship. When one considers, however, the enormous complications and multiplicity of detail in the oil-engine installation, it would seem as if Mr. Orde's figures are not far out. In the largest oil-engine ship now at sea, as Mr. H. F. Fullager pointed out, there are in the engine room, including auxiliary engines, 24 working cylinders, 96 cylinder valves, 160 cams, innumerable levers, joints and tappets, and nine compressors, in five of which the compression reaches 900 pounds per square inch, so that there is, at least, much to keep in good repair.

Mr. Orde, in the machinery adopted by him for the equipment of the type ship, preferred two-stroke cycle engines driving twin screws, and here, again, he was criticised, the contention being that, first, the single screws gave better results, and that the four-stroke cycle engine was preferable. Mr. Knudsen, for instance, stated that the oil consumption in the two-stroke cycle engine was 20 per cent. higher than in the four-stroke cycle engine. Mr. G. Zulver, of the Anglo-Saxon Petroleum Company, contended that the fuel consumption, as ascertained from the working of the *Valcanus*, was 15 per cent. less, although the capital charges were 10 per cent. higher, and, further, that the wear and tear as ascertained from the *Vulcanus* should be £50, as compared with £130 per month for a steamer. There can be no doubt, however, that the two-stroke cycle engine must be the ultimate design if powers are to be increased to any considerable extent; for while the four-stroke cycle engine may be commended for the present because it affords less complications and less risk of breakdown, it nevertheless requires a greater number of cylinders and a greater weight for the same power.

In summing up the working costs independently of fuel consumption it is found that there is not much difference in the three types of engines of which particulars are given: the triple-expansion steam engine, the geared turbine, and the Diesel engine. Insurance of oil engines was put a little higher in these comparisons than it ought perhaps to have been, because, while Lloyd's underwriters were at the outset inclined to look with doubt upon the oil engine, more confidence now obtains, and it is probable that insurance may not much exceed that of the steam engine. Including insurance, wages and provisions, wear and tear, stores, port charges, management, and 5 per cent. for depreciation, with a like allowance for interest on capital, the triple-expansion engine of the type ship, with superheater, works out at £30,720 per annum. The Diesel engine, accepting the insurance at about the same total, costs about £31,800 per annum, and the geared turbine, using superheated steam, at £29,970. Thus the approximate cost of working a ship per ton of freight-earning cargo carried is for the triple-expansion engine 57.5d. per annum; for the Diesel engine, 57.7d.; and for the geared turbine with reciprocating engine, 55d.

It must not be forgotten, however, that there are problems in mechanics yet to be overcome in connection with the application of the oil engine. Sir Charles Parsons referred to some of these when dealing with the severity of the treatment of the cylinder due to the alternate heating and cooling of the walls, because of the difficulty of preventing the metal from cracking. Hopkinson and David, as he stated, have shown by experiments that over the periods of maximum temperature and pressure the flame is very transparent, and that the heat of the cylinder walls increases with the depth of the flame and also with the density. Some of the experiments show that if the size and density are each doubled, the heating of the walls is also approximately doubled. Consequently the larger the cylinder the more severe is the heat; further, the larger the cylinder the slower should be the revolutions for a given piston speed, and the longer the duration of each stroke, and this again increases the

effect of the heat on the walls of the cylinder. Thus the increase in size and power of oil-engine cylinders must necessarily increase the difficulties from the metallurgical point of view as well as augment the capital cost per unit of power without affecting materially the economy. With turbines, on the other hand, increased size reduces cost.

It follows that under present conditions the largest field for the oil engine is in relatively small-powered ships, and that the commercial condition is best met on those routes where oil is easily obtained from wells in adjacent countries. This, it may be said, is the principal deduction from the comprehensive discussion at the Newcastle Institution. Each type of engine has its sphere, and good will accrue from recognition of the limitations, because advocates who assume that any one type is universally applicable must necessarily provoke opposition. The mechanical problems are becoming more and more understood. There may not be a greater display of confidence than formerly in the potentialities of the oil engine, but a free discussion of difficulties in the light of experience must have its effect upon all workers in this field of engineering. The one warning necessary is against hasty assumption and premature attempts at high powers. We echo the hope expressed by Mr. Tom Westgarth, that progress will not be delayed by too much hurry in the first instance.—"Engineering."

SHIPS.

UNITED STATES.

ELECTRIC DRIVE ON COLLIER.

THE "JUPITER," BUILDING AT MARE ISLAND, WILL BE EQUIPPED WITH ELECTRIC PROPELLING MACHINERY.

DESCRIPTION OF THE DRIVE.

The *Jupiter* is one of the new Government colliers and is a sister ship to the *Neptune* and *Cyclops*. The *Cyclops* is equipped with reciprocating engines and the *Neptune* with a turbine drive connected to the propellers by gearing. These ships have a displacement of about 20,000 tons and carry something like 12,000 tons cargo. They are designed to operate at 14 knots. The *Cyclops* in her 48-hour trial averaged 14.6 knots, with a coal consumption for the main engines only of 1.485 pounds per I.H.P. per hour, the total I.H.P. of both engines in this run being 6,705. The average speed of propellers was 92 r.p.m. The results of trials of the *Neptune* cannot be given here, but the fact has been published that her performance was not as good as that of the *Cyclops*, the difference being attributed to less efficient propellers and to inefficiency in the turbine. It is reported that the gearing works in an entirely satisfactory manner.

The following table shows comparisons of the known data concerning the equipments of these three vessels:

	Cyclops.	*Jupiter.*	*Neptune.*
Displacement, tons	20,000	20,000	20,000
I.H.P., 14 knots	5,600	.....	.....
Engine or turbine speed, 14 knots, r.p.m.	88	2,000	1,250
Propeller r.p.m., 14 knots	88	110	135
Weight driving machinery, tons	280	156	.....
Steam consumption in lbs. per S.H.P. hr.	†14	*12	.....

Character driving machinery—*Cyclops:* Two triple-expansion engines.
Character driving machinery—*Jupiter:* One turbo-generator and two motors.
Character driving machinery—*Neptune:* Two turbines each with gearing.

† Estimated. * Tested.

The collier *Jupiter*, which is being built at the Mare Island Navy Yard, is to be equipped with electric propelling machinery designed by W. L. R. Emmet and built by the General Electric Co. The generating unit consists of a 6-stage Curtis turbine connected to a bipolar alternator, the speed of this unit at 14 knots being about 2,000 r.p.m. and the voltage about 2,200. This generating unit delivers electricity for two motors, one mounted directly upon each propeller shaft. These motors have 36 poles, therefore the ratio of synchronous speed reduction is 18 to 1, the propellers at 14 knots being designed to operate at 110 r.p.m. In addition to this apparatus there is a switchboard equipment which provides oil switches for connecting the motors for either direction of rotation and instruments which show and record the electric power delivered to the motors. There are also two water-cooled resistance devices which are put in circuit with the revolving part of the motors in the process of reversing. Connections for the insertion of these resistances are made by sliders on the motor shafts operated by levers attached to the motor frames.

The generating unit and motors are self-lubricating and self-ventilating.

Sheet-metal ducts will be connected to their air outlets in such a manner that the heated air will be led to the suction of the fireroom blowers so that it will not be released in the engine room.

SAFETY INTERLOCKS.

Since in such an equipment it is only necessary to make such electricity as is required for the actual driving of the ship, it is possible to so design the equipment that the maximum power which can be delivered by the generator is not greatly in excess of that of the regular requirements, and this fact practically overcomes the possibility of destructive trouble through wrong connections. It has, however, been thought desirable to arrange interlocks in such a manner that wrong connections cannot be made, the conditions being such that these interlocks involve no complication or uncertainty. Provision is thus made by which the go ahead switch and reversing switch cannot be closed at the same time and by which neither switch can be closed unless the resistance is in circuit with the motor secondary. For the levers which operate the connection and disconnection of the resistance, magnetic locks are attached which are energized by the field circuit of the generator. These locks prevent the movement of these levers until the generator has lost its field, and thus prevent any possibility of burning of contacts through movements at the wrong time or in the improper manner.

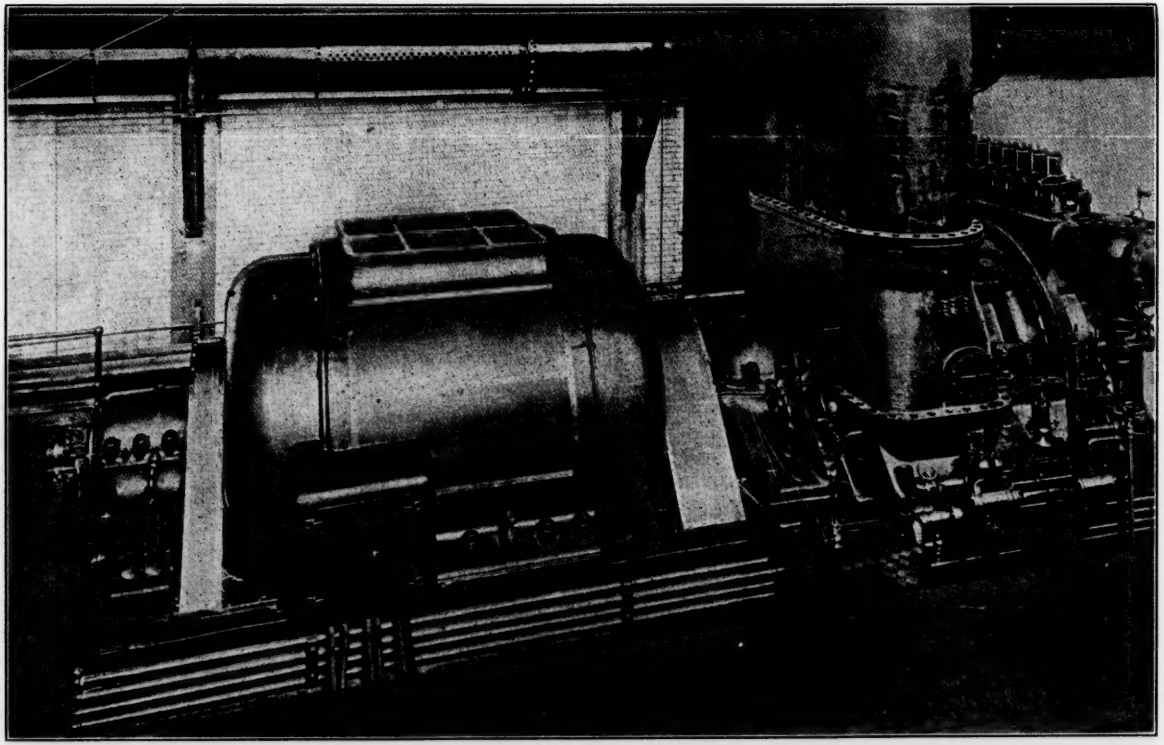
SPEED CONTROL.

The speed of the motors in this vessel will be changed by variations in the speed of the generating unit, the ratio of speed reduction remaining fixed. The changes of speed, however, are not made by throttling, as is usual with ship turbines. The turbine is equipped with a governor of novel construction, which is so arranged that it is capable of automatically holding the speed at any point from about 5 knots up to the maximum. The setting of this governor is accomplished by the movement of a fulcrum which is controlled from a point near the switchboard, and operating levers. Thus the operating engineer, without changing his position, can run the vessel at any desired speed ahead or astern, can stop and start, and from his instruments can see the speed and amount of power delivered to each propeller. When the apparatus is installed in the ship, arrangements will also be made by which in the same position he can open and close the main throttle valve by hand or can trip it so that it closes instantly. The generating unit is also equipped with a simple automatic device entirely separate from the governor, which trips the main throttle in case the speed of turbine exceeds a certain predetermined limit.

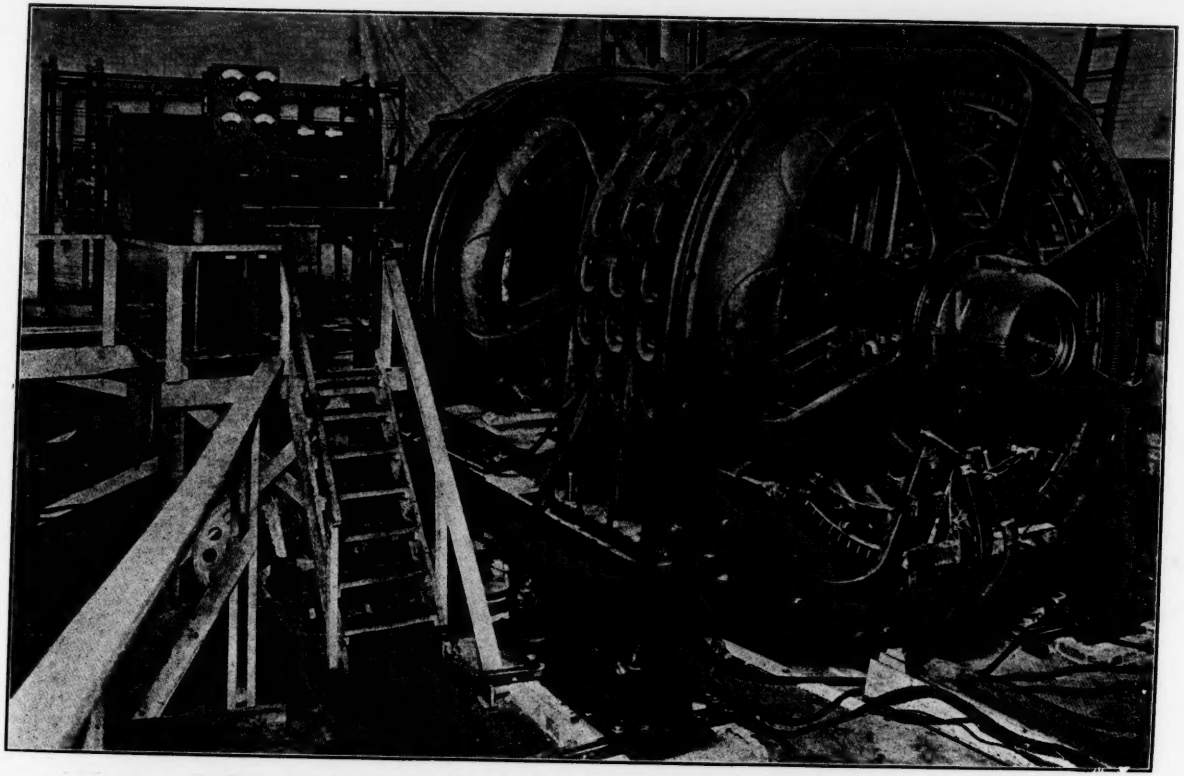
ELECTRICAL CONDITIONS.

The electrical methods used on this ship constitute the simplest known form of electric power transmission, and apparatus of similar character is used for a great variety of purposes on shore under conditions far more complicated and difficult and with an immunity from trouble which is practically complete. In many years of production of alternating apparatus of this general character and voltage there has hardly been a case of insulation failure recorded, and an investigation of this fact will convince any one that the elecrical part of this equipment is particularly reliable and that in this respect it is far superior to the types of apparatus which have been used for ship propulsion. Furthermore, its nature is such that, in the event of trouble, it could be very easily and quickly repaired. The switching apparatus is of an entirely reliable standard type, it can be easily replaced or repaired if it should get out of order, and if it were all removed the ship could be easily operated with temporary connections.

The turbine is so designed that all of its parts are accessible and re-

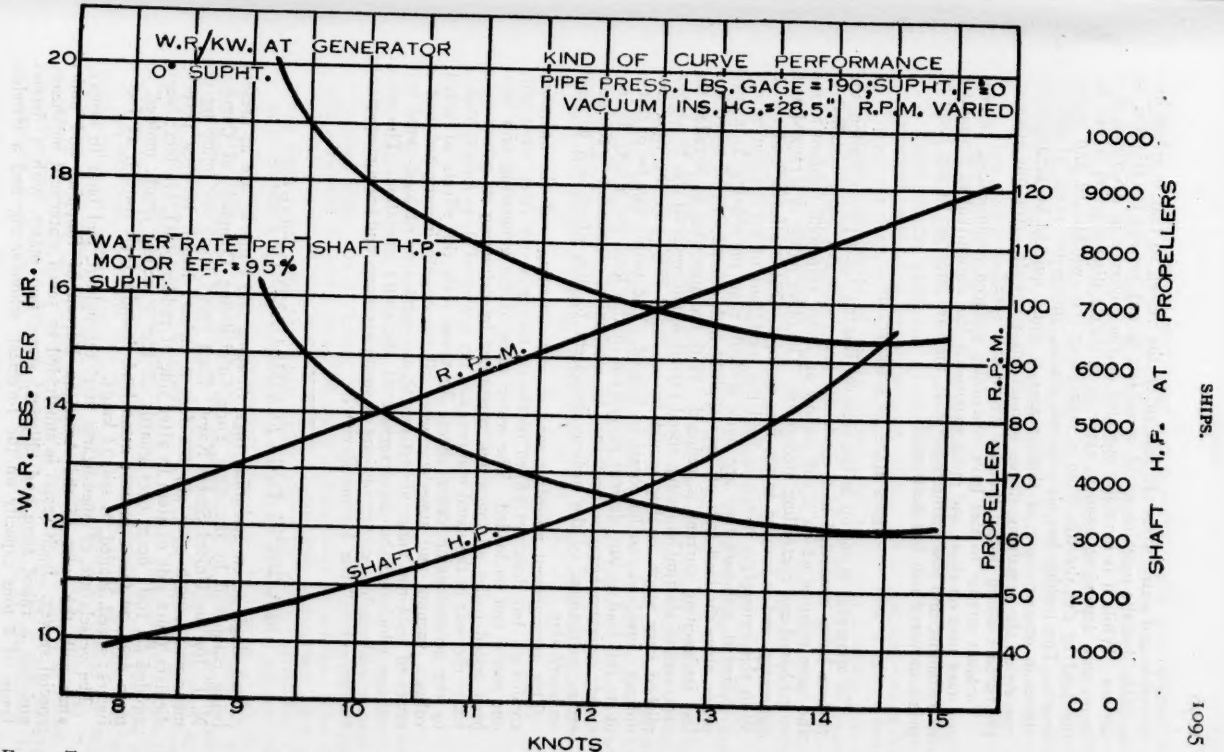


GENERATING UNIT, UNITED STATES COLLIER "JUPITER."



TYPE 1-36-2,750-110-2,300 VOLT MOTORS, UNITED STATES COLLIER "JUPITER."

W.R/KW. AT GENERATOR



From Test of Generating Unit Run Under Proper Conditions of Speed, Load and Voltage and Reduced by Known Efficiency of Motors.

placeable and extra parts are carried on the vessel, so that it seems impossible that the machine could be subject to any serious interruption of service. While it is normally operated on the governor as described, it is in no way dependent upon it, since by a simple disconnection it can be operated by the throttle from a point near the switchboard and motor levers. The turbine has six separate stages, and in the event of loss of blades in one or more of these stages it could quickly be arranged to operate on the remaining stages without any renewals. Any one stage of the turbine would propel the ship above half speed.

Turbines similar to this have practically replaced reciprocating engines for most uses on shore and much experience has shown that in the matter of reliability and low maintenance cost, they are superior to any form of prime mover which has been used.

TESTS MADE IN SCHENECTADY.

The apparatus is set up in the General Electric Co.'s power station, the turbine being connected to a condenser and one motor being installed in the same position which it will occupy on board ship with relation to the switchboard and controlling mechanism. The other motor is arranged as a generator, and is coupled to the first motor so that it affords load for it. With this generating motor as load the conditions of service can be approximated, although they are more difficult than the driving of a propeller, since the load falls off only slightly with diminutions of speed unless its exciting current is reduced. With the apparatus so installed the processes of starting, stopping, speed variation, and reversal can be accomplished very much as they would be on board ship, and the time and difficulty involved can be correctly judged. The motor can be operated at its full load or any desired proportion of this load and at any desired speed, variation being accomplished by mechanical connection to the governing mechanism of the turbine equivalent to that which will be used on board ship.

This arrangement affords means of experimenting with the operating conditions, but does not afford means of testing water rates, since only one motor can be loaded. The water rates of the generating unit have been tested under all loads and conditions by delivering its power to a water rheostat in the usual manner, and the record of these tests is shown by the accompanying curve sheet. In these tests the effects of speed, voltage, vacuum and superheat were all thoroughly investigated and a series of tests run under almost exactly the conditions of load, speed and voltage which will be characteristic of the ship's operation. The tests were made under exact conditions of steam pressures and vacuum specified for the ship.—"The Marine Review."

SEA-GOING TUGS *SONOMA* AND *ONTARIO*.

Two single-screw sea-going tugs, *Sonoma* and *Ontario*, are at present being completed by the New York Shipbuilding Company, at Camden, N. J., for the United States Navy. Each vessel is of the following dimensions: Length over all, 185 feet 2 inches; length between perpendiculars (fore side of stem to after side of rudder post), 175 feet; beam, molded, 34 feet; beam over guards, 35 feet 6 inches; depth, molded, to main deck, 20 feet 3 inches; mean load draught (salt water), 12 feet 6 inches; speed, loaded, at sea, 14 knots.

The vessels are of open-hearth steel throughout, and built in accordance with the rules of the American Bureau of Shipping and under special survey. Each vessel is subdivided by seven watertight bulkheads, and has a raked stem and elliptical stern, two masts, with a derrick boom of 5 tons capacity on the foremast, schooner rig and a wireless

outfit. These tugs are very large and powerful, being well adapted for the work they would be called upon to do in the event of war.

Quarters for a naval complement of four officers, three chief petty officers and thirty-five men are provided for.

There is fitted forward a steam windlass, combined with a towing bitt, the windlass having warping ends. A reversible capstan is fitted on the main deck aft. Two steam winches of 5 tons capacity each are located, one at the foremast on the main deck, the other at the main mast on top of the deck house. A towing machine is provided in the after end of the main deck house and supplied with necessary hawser rollers, etc.; a towing bitt is also fitted aft.

The steam-steering engine is located in the engine room, and is connected with the pilot house; a combined steam and hand-steering wheel is fitted on the after end of the deck-house top. The vessel's boats consist of one 21-foot motor dory, one 28-foot whale boat and one 16-foot dinghy, the former two being slung under davits. One 4-inch Monitor fire-hose nozzle is fitted on top of the pilot house and a similar one on top of the deck house aft. A machine shop, amply equipped, is located upon a platform on the port side of the engine room.

A double bottom and feed-water tanks are fitted under the boilers. Peak tanks are arranged for the carrying of fresh or salt water. The coal bunkers are abreast of the boiler room, and a cross-bunker is aft of the boiler room. The drinking-water tanks contain 4,000 gallons of fresh water, and there is a gravity tank of 500 gallons. Refrigerator space is provided with a capacity of 550 cubic feet.

The propelling machinery is placed amidships, and consists of two single-ended Scotch boilers, arranged fore and aft, with the fireroom between the two. They are 16 feet in diameter and 11 feet between heads, with a working pressure of 200 pounds per square inch. The main engine is of the vertical, inverted-cylinder, triple-expansion, surface-condensing type, of about 1,800 horsepower, with cylinders 19¾, 31½ and 54¼ inches diameter, respectively, with a common stroke of 36 inches.

ENGLAND.

H. M. S. *CONQUEROR.*

The *Conqueror* on her trials attained a mean speed of 22.126 knots, whereas the sister ship—the *Thunderer*—got 20.08 knots; the *Monarch*, 21.88 knots; and the *Orion*, 21.02 knots. The speed aimed at in these various ships was 21 knots, but the propellers are different, and there is a reserve of steam power to meet contingencies in service. The designed power of the machinery was 27,000 shaft horsepower, and on the eight hours' trial the actual mean was 28,430 shaft horsepower, the two starboard turbines developing 14,380 shaft horsepower, and the two port turbines 14,050 shaft horsepower, the mean revolutions of all four shafts being 322 revolutions per minute. The steam pressure at the boilers was 230 pounds, and at the engines 217 pounds, the vacuum being 28.3 inches, with a barometer reading of 29.82 inches. The coal consumption throughout the trial was 1.74 pounds, which is very satisfactory.

On the 30-hours' coal-consumption trial the mean power developed was 19,081 shaft horsepower, almost exactly equally divided between the two starboard and the two port shafts. The boiler pressure was 226 pounds, the vacuum 28.6 inches, with a barometric reading of 29.57 inches. Six runs were made on this trial, and the mean speed was 19.36 knots. Equally with this and the full-power trial the ship was loaded to her service draught, and the stokeholds were under practically natural-draught conditions, no difficulty being experienced in maintaining full

steam pressure. So there is no doubt that, so far as the design is concerned, the machinery is highly satisfactory.

There are eighteen boilers in the ship, of the Babcock & Wilcox type, and the ratio of heating surface to grate area is 35:1, while 2¼ square feet of heating surface is provided per unit of power. The turbines are of the Parsons type, mounted on four shafts, there being two high-pressure ahead turbines and two low-pressure ahead turbines, while the astern turbines are also arranged similarly in series, there being two high-pressure and two low-pressure.

The *Conqueror*, which is of 22,500 tons displacement at a draught of 27 feet 6 inches, has a length between perpendiculars of 545 feet, a beam of 88 feet 6 inches, and a depth molded of 43 feet 6 inches. There are ten 13.5-inch guns, placed in pairs in barbettes, so arranged on the center line that all can be fired on either broadside, and, in addition, there is an auxiliary armament having a combination of 6-inch and 4-inch guns. The *Conqueror* has been built and engined by Messrs. William Beardmore & Co., the keel having been laid on April 5, 1910.—"Engineering."

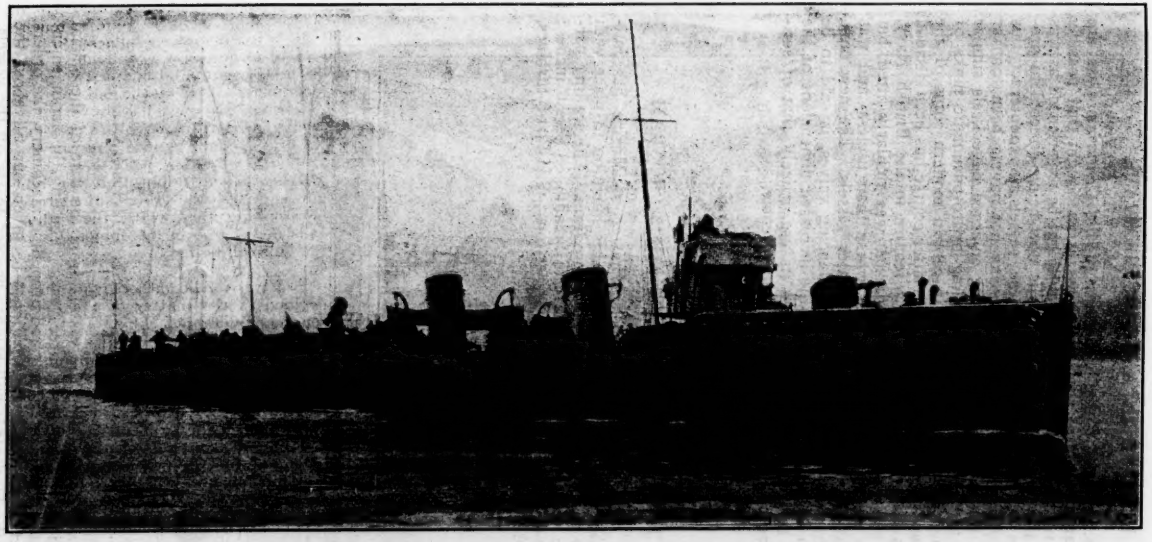
NAVY ESTIMATES, 1912-13.

The British Navy Estimates for 1912-13 provide for a sum of £13,971,527 being expended on shipbuilding (excluding repair work), as compared with £15,063,877 for 1911-12. Of this total £12,067,727 will be spent on the continuation of work at the ships already under construction, and £1,903,800 on commencing the ships of the new program. In last year's Estimates the corresponding figures were £13,325,232 and £1,738,645 respectively.

The new vessels to be laid down include four large armored ships (as compared with five last year), eight light armored cruisers, twenty destroyers, a number of submarines, a coastguard cruiser, and other subsidiary vessels. One of the large armored ships is to be built at Portsmouth Dockyard and the other at Devonport Dockyard, the provision for each being slightly over £166,000. The other two large armored ships will be built by contract. For one the expenditure during the present financial year is estimated at only £2,814, and for the other at £59,789. Presumably the two Dockyard-built vessels and one of the contract-built vessels will be battleships, while the other will be a battle cruiser of the improved *Lion* class.

Of the eight light armored cruisers, one each will be commenced at the Chatham and Pembroke Dockyards, and an expenditure of nearly £113,000 is provided for on each. The remaining six vessels of this class will be constructed by private firms, and for them the provision is slightly over £12,000 each. These cruisers, which will represent an entirely new type, will be smaller than the second-class protected cruisers of last year's program, and indeed they will be the smallest, cheapest and fastest vessels protected by vertical armor ever proposed for the British Navy. In the matter of speed they will be swift enough to overhaul any torpedo craft, so that they will act as destroyers of destroyers.

For the twenty destroyers to be laid down under the new program a sum of £1,053,711 has been set aside, and sixteen of the vessels have already been provisionally ordered—four from the Fairfield Shipbuilding & Engineering Co., Govan, and two each from Messrs. William Denny & Brothers, Dumbarton; Messrs. Yarrow & Co., Scotstoun; Messrs. Swan, Hunter, and Wigham Richardson, Wallsend; the Parsons Marine Steam Turbine Co., Wallsend; Messrs. John I. Thornycroft & Co., Southampton, and Messrs. J. Samuel White & Co., East Cowes. On the submarines to be commenced at Chatham Dockyard the expenditure allotted for this year is £36,376, while for the similar vessels to be built in private yards £49,838 has been provided. A sum of £9,153 is apportioned for the new coastguard cruiser, and a total of £10,204 for two small tugs.



H. M. TORPEDO-BOAT DESTROYER "*FIREDRAKE*."

H. M. S. *Firedrake*, one of the three special torpedo-boat destroyers ordered by the British Admiralty last year to be designed and constructed by Yarrow & Co., Limited, Glasgow, has recently completed a satisfactory full-speed trial. The type to which this vessel belongs comprises also H. M. SS. *Lurcher* and *Oak*, and all three are 255 feet long, with a beam of 15 feet 7 inches.

The *Firedrake* is propelled by Parsons turbines driving twin screws, steam being supplied by three of the latest type of Yarrow boilers fitted with the firm's feed-heating device and arranged for burning oil fuel. During the eight hours' full-speed trial at the mouth of the Clyde the vessel maintained a mean speed of 33.17 knots, the contract speed being 32 knots. The mean horsepower during the full-speed trial was slightly over 19,000. The displacement on trial was about 780 tons.

The *Firedrake* type is built to maintain a high speed against heavy seas, having a high forecastle forward. From the results obtained the *Firedrake* type is considered a highly satisfactory form of destroyer, and a distinct advance in this class of vessel.—"The Engineer."

ROLLING IN A HEAVY SEAWAY.

As the characteristics of the battleships of the *Centurion* class are somewhat similar to those of the *Orion* class, it is thought that possibly the same excessive rolling in a heavy seaway might be experienced as was recently felt in the *Orion;* hence attention is now being directed to providing additions to the equipment of these vessels to counteract this tendency. The *King George V* and *Centurion* not having been too far advanced, there was a choice between fitting enlarged bilge keels similar to those added to the *Orion,* and anti-rolling hydro-pneumatic tanks of new design which have proved effective in certain modern ships of the mercantile marine. We understand that the latter course has been decided upon. These tanks are not exactly a new feature in the British Navy, as in the early eighties the turret ship *Inflexible* was similarly fitted, but owing to faulty design the device proved worse than useless, and was discarded in that ship and not repeated in later vessels. The new tank, however, owing to its efficient system of balancing valves, is said to be very effective in its action, and owing to the increased dimensions of the *Centurion* class the objection urged against its use, namely, that of seriously encroaching on available space, can be overcome.

FRANCE.

FRENCH BATTLESHIPS OF THE 1912 PROGRAM.

According to "Le Yacht," the French Minister of Marine has intimated to the Dockyards at Brest and Lorient his intention to place with each one of the two battleships of the 1912 program. The leading particulars of these vessels are: Length, 541 feet; breadth, 88 feet 6 inches; draught, 29 feet 6 inches; displacement, 24,000 tons; S.H.P. of turbines, 28,000; speed, 20 knots.

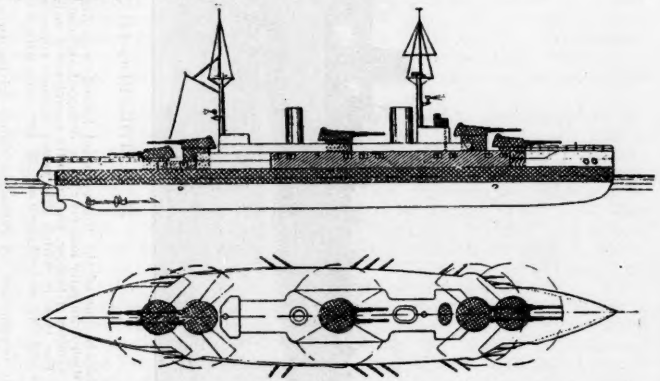


FIG. 1.—FRENCH BATTLESHIPS OF THE 1912 PROGRAM.

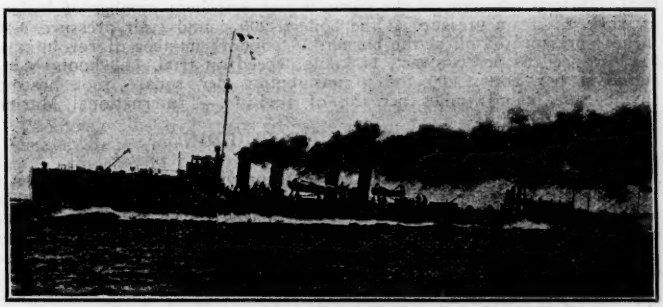
The time allowed for building the vessels from the date of the official orders to the official trials is to be three years. Pending the arrangement of contracts for the armament, armor and propelling machinery, the orders for the construction of these ships will not be given out till about the 1st of May, but the Dockyards will at once proceed to lay off the vessels,

so that they may be in a position to commence the ships directly the order is received.

Some further particulars are given in an article by M. Henri Bernay in the same journal, from which Fig. 1 is taken. The main armament will consist of ten 13.4-inch guns, placed in pairs in five turrets, all situated on the center-line of the ship. In this respect the new vessels differ from the *Jean Bart* and *Courbet,* now building, the latter vessels having twelve 12-inch guns placed in six turrets, of which two are placed, one on each side, near amidships, with an arc of fire of 180 degrees. In the 1912 ships the guns in the center turret will each have an arc of fire on each side of 120 degrees, or 240 degrees in all. Compared with the *Jean Bart* class, therefore, an equal number of guns can be trained for broadside fire, while the number which can be trained for bow or stern fire is reduced by one-half. The secondary armament will consist of twenty-two 5.5-inch guns, and there will be four 17.7-inch torpedo tubes. The main armor belt will be 10.6 inches thick and extend 8 feet 3 inches below the water line and about the same distance above. Towards the ends of the ships the surface protected by armor is somewhat less in extent than in the *Jean Bart* and *Courbet.* The armor protecting the secondary batteries will be 7 inches thick. A third battleship of this class is to be ordered in August to replace the *Liberté.*—"The Shipbuilder."

FRENCH DESTROYER *DAGUE.*

This destroyer has the following dimensions: Length over all, 255 feet 11 inches; length between perpendiculars, 251 feet 1 inch; beam, 26 feet 1 inch; beam at load waterline, 25 feet 10 inches; depth, 16 feet 9 inches; draught at the stern, 9 feet 8 inches; trial displacement, 730 tons; full load displacement, 770 tons.



NEW FRENCH DESTROYER "DAGUE."

The hull was designed by the Gironde Works, and the turbines supplied by the Brequet firm, which is one of the two builders of marine turbines under French patents. The hull is divided into ten watertight compartments. She has a high freeboard forward, where there is a forecastle deck, which insures better seaworthy qualities than previous types of the destroyers built for the French Navy.

Steam is supplied to the main and auxiliary engines by four du Temple water-tube boilers, the heating surface of the boilers being 10,668 square feet. Liquid fuel is used, and each boiler has eleven Thornycroft burners. They deliver the fuel under a pressure of 143 pounds per square inch.

Thornycroft fuel heaters are also used. The working pressure of the boilers is 173 pounds.

The main engines consist of two independent sets of Brequet turbines, each driving a separate shaft. In each casing are located an ahead and an astern turbine. They have been designed to develop an average of 1,500 shaft horsepower at a speed of 630 revolutions per minute. The contracted speed of the vessel was 31 knots, but during the trials a speed of 33.118 knots was attained. The Brequet turbines are of the impulse type. The power of the steam acting on the blades is balanced by the propeller thrust, any difference between the two being taken up by a small thrust bearing. Each turbine drives a single three-bladed propeller, 7 feet 1 inch in diameter and 6 feet 7 inches pitch.

Each turbine has its own independent condenser, with a total surface of 27,340 square feet. The fuel is carried in the double bottom and longitudinal tanks on both sides of the boiler rooms. The total capacity of the tanks is 5,700 cubic feet, which gives a theoretical steaming radius of 2,060 miles.

The armament consists of two quick-firing 4-inch guns located forward and aft and four 18-inch torpedo tubes.

RESULTS OF OFFICIAL TRIALS.

Six Hours' Duration Trial.—Displacement before trial, 733 tons; draught at the stern, 9 feet 8 inches; number of boilers at work, 4; steam pressure at the boilers, 214 pounds; steam pressure at the turbines, 200 pounds; air pressure, 7.6 inches; oil pressure at the burners, 142 pounds; number of revolutions, 667; contract speed, 31 knots; mean speed for six hours, 33.118 knots; oil consumption as per contract, 12.50 tons; oil consumption on trials, 11.15 tons; consumption per square foot of heating surface, 1.22 pounds; miles per ton of fuel, 2.83.

Eight Hours' Consumption Trial at 14 Knots.—Displacement before trial, 734 tons; draught at the stern, 9 feet 8 inches; number of boilers at work, 2; steam pressure at the boilers, 200 pounds; air pressure, 1.16 inches; pressure of oil at the burner, 58 pounds; number of revolutions, 244.16; speed as per contract, 14 knots; speed on trial, 14.3 knots; consumption per hour, 1.03 tons; consumption per square foot heating surface, .02 pound; miles per ton of fuel, 14.—"International Marine Engineering."

FRENCH WARSHIP BUILDING.

The Navy Program of France for 1913 provides for the laying down of four battleships, in addition to the five to be advanced and the two to be completed during next year. This makes eleven in all in process of construction. The two to be completed are the *Jean Bart* and the *Courbet*, which were launched respectively at Brest and Lorient in September last. These vessels are of 540 feet in length, 88½ feet in beam, and at 29 feet draught displace 23,100 tons. The Parsons turbine machinery, of 28,000 shaft horsepower, is to give them a speed of 20 knots. The five battleships now in progress are the *Bretagne* and *Provence*, building at Brest and Lorient, and the *France*, *Paris* and *Lorraine*, building by contract, the first at St. Nazaire and the second at La Seyne. These are to be generally of the same dimensions as the *Jean Bart*, but 500 tons more is allowed for displacement on the same draught, and, in consequence, the power of the Parsons turbines will be 29,250 shaft horsepower, to give 20 knots speed. Of the four new ships to be laid down, two will be commenced in the Government dockyards on May 1, next year, and two at Brest and Lorient respectively on January 1, 1914. Although no official particulars are given regarding these vessels, it is understood that they will be larger than those now in progress, the displacement tonnage being about 25,500 tons, which will involve a further increase in power to ensure the same speed. As regards the armor,

the thickness of the main broadside belt in the *Jean Bart* is 11 inches, tapering fore and aft, with 7-inch armor up to the upper deck, while the bulkheads are also of 7-inch armor. The gun barbettes are of 10½-inch armor, and the secondary guns are placed on the broadside, behind the 7-inch upper belt. In the *Jean Bart* the armament includes twelve guns of 12-inch caliber, twenty-two of 5.5-inch caliber, and four 3-pounders. In the five later ships of the *Bretagne* type there will be ten 13.39-inch guns, twenty-two 5.5-inch, and four of 1.85-inch bore. There will be four submerged torpedo tubes. The complement of these ships is to be 34 officers and 1,075 men. So far as the ships to be laid down next year are concerned, the prospects are in favor of corresponding armament, the 13.39-inch being regarded as sufficient.

No cruisers are provided for, and as regards the torpedo-boat destroyer program, nine vessels are to be completed next year, seven of them at private works and two in the dockyards. Three will be "continued," and three will be laid down. The length of the later vessels in progress is 265 feet 9 inches, the beam being 25 feet 7 inches, and at 9 feet 10 inches draught the displacement is 850 tons. These are twin-screw turbine-driven ships, and the armament includes two 3.9-inch and four 2.56-inch guns. Of submarine boats, eight will be completed, ten continued, and three will be commenced. The submarines are steadily increasing in size. These latter vessels are to be 242 feet 9 inches long and of 19 feet 8 inches breadth. Internal-combustion engines of 4,800 horsepower are to be fitted, in order that the boats may have a speed on the surface of 20 knots, a fact itself illustrative of the great advance made in this type of craft. There will be eight tubes for launching torpedoes, and the crew will include three officers and thirty-seven men.

Certain miscellaneous ships are included. There is in progress a submarine-mine-laying ship called the *Cerbere*, of 566 tons, and with machinery of 6,000 horsepower, to give a speed of 20 knots. The vessel will carry one 2.95-inch gun, and will be manned by a crew of three officers and 62 men. This vessel was put in hand at St. Nazaire on August 5 of last year. A transport named the *Seine* is also in progress, the displacement being 3,160 tons and the speed 12 knots. Next year there is to be laid down a gunboat to be called the *Balny*, of 214 tons, to carry six 1.46-inch guns. Another special ship to be laid down is a despatch vessel, which will have a displacement of from 1,600 to 1,800 tons, with a speed of 17 knots. She will carry two 5.5-inch and six 2.95-inch guns. The "Moniteur de la Flotte," from which we have taken these particulars, states that the total credit for new construction will amount to nearly 7¼ millions sterling for direct expenses, and 360,000*l.* for miscellaneous expenses.—"Engineering."

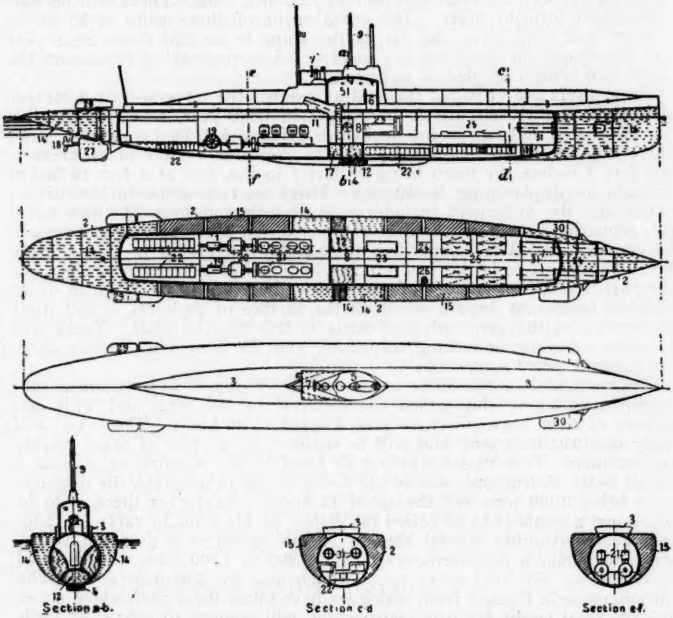
ITALY.

THE ITALIAN SUBMERSIBLE *ATROPO*.

In the Italian naval construction program for 1910-11 provision was made for building a flotilla of submarines, the units of which were to be of a variety of types, in order to test the relative merits of different designs. Two of these vessels, the *Galileo Ferraris* and the *Giacinto Pullino*, are of the "Cavallini" type, and are nearing completion at Spezia. They have a displacement of 400 tons, a speed of 18 knots above water and a speed of 14 knots when running submerged. Two other boats, the *Nautilus* and the *Nereide*, are being built at Venice, and are of the "Bernandis" design, while eight units, the *Argo*, *Fisalia*, *Medusa*, *Salpa*, *Velella*, *Yalea*, *Yantina* and *Zoea*, are being constructed by the Fiat Company. The latter eight vessels are very similar in type to the Laurenti submarine *Foca*, except that they are somewhat larger. Their

displacement is 250 tons at the surface and 300 tons when submerged, and their corresponding speeds 13 and 8½ knots.

The thirteenth and last vessel of the new flotilla was built by Messrs. Krupp and was launched at Kiel in the end of March. This boat, the *Atropo*, is illustrated in the accompanying engraving. She has a length between perpendiculars of 146 feet and a molded breadth of 14 feet 6 inches, her underwater displacement being 320 tons. On the surface

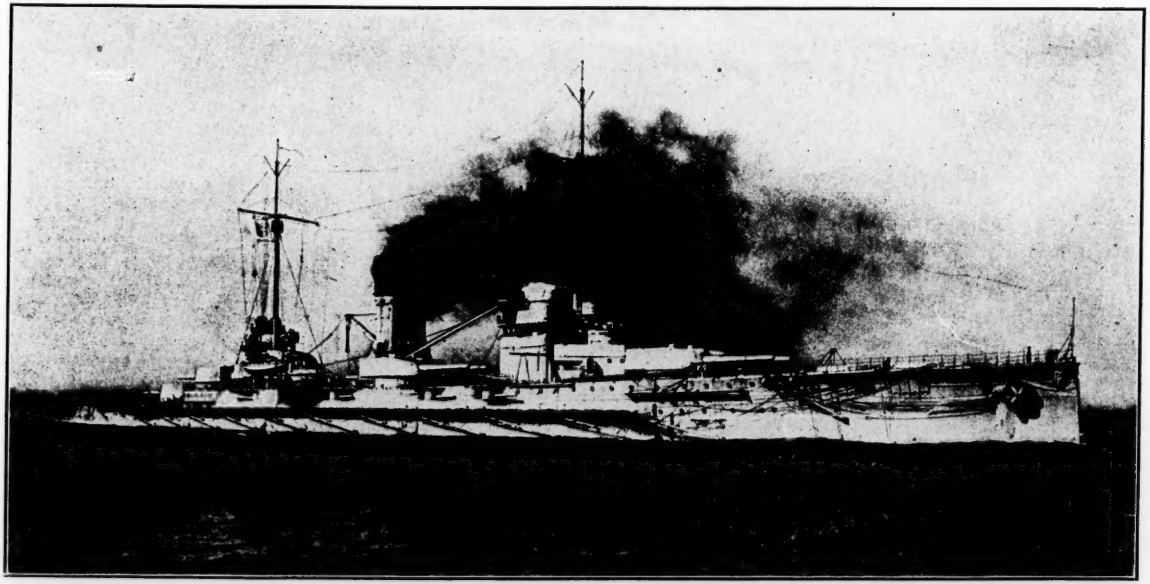


1. Inside or watertight hull.
2. Outside hull.
3. Surface navigation platform.
4. Detachable safety weight.
5. Conning tower.
6. Steering gear for submarine navigation.
7. Steering gear for surface navigation.
8. Diving rudders control.
9. Periscopes.
10. Signal mast.
11. Torpedo passage.
12. Water ballast for inside hull.
13. Trimming tanks.
14. Water ballast for outside hull.
15. Oil-fuel tanks.
16. Lubricating-oil tanks.
17. Fresh-water tanks
18. Reversible propellers.
19. Propeller reversing gear.
20. Electric motors.
21. Diesel engines.
22. Storage batteries.
23. Officers' quarters.
24. Galley.
25. Crew's quarters.
26. Closet.
27. Lower vertical rudder.
28. Upper vertical rudder.
29. After diving rudders.
30. Forward diving rudders.
31. Torpedo tubes.

THE ITALIAN SUBMERSIBLE "ATROPO."

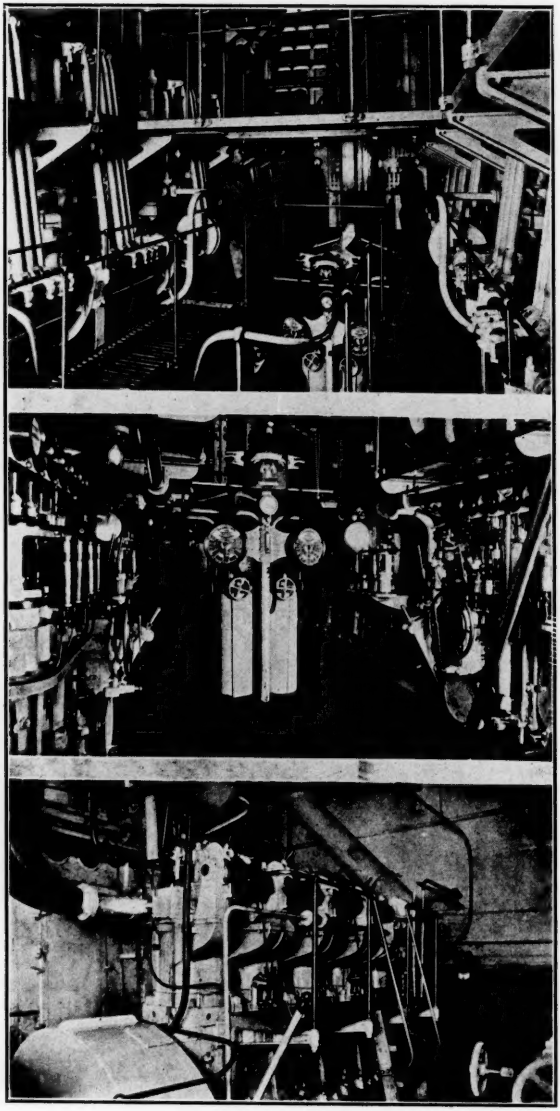
her speed is 12½ knots, and in this condition propulsion is secured by two two-cycle Diesel engines of 350 horsepower each. When running submerged she employs two electric motors of 200 horsepower each, and attains a speed of 8 knots. The radius of action is 1,300 miles above water and 40 miles submerged.

Heavy-oil Diesel engines are being used in all the vessels of this flotilla, those of the *Atropo* being of the Germania design, and those of the others either M.A.N. or Fiat productions.

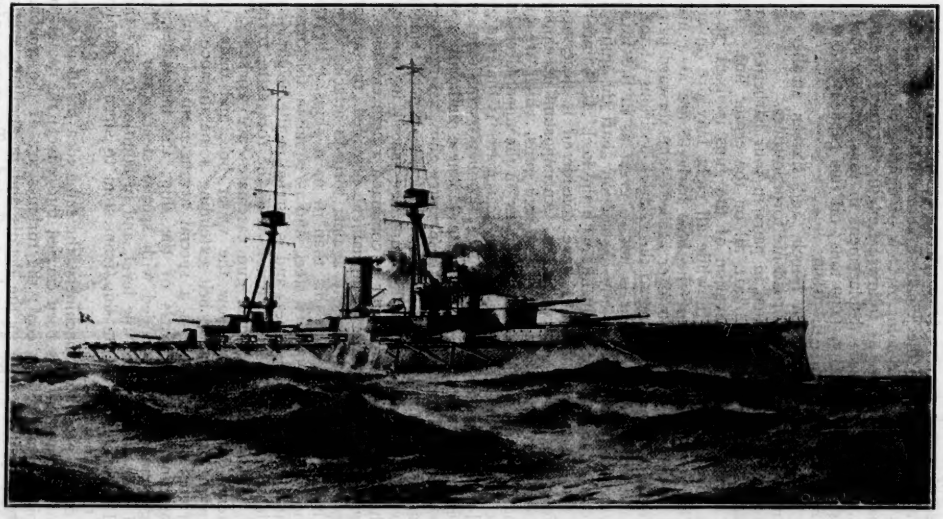


Length, 590 feet. Beam, 88 feet. Displacement, 23,500 tons. Maximum coal supply, 2,500 tons. Speed, 29 knots.
Armament: ten 11-inch, 50-caliber guns; twelve 6-inch rapid-fire guns. Torpedo tubes, 4 submerged.

THE GERMAN TWENTY-NINE KNOT BATTLE-CRUISER "MOLTKE."



VIEWS IN THE ENGINE ROOM OF THE "SELANDIA." (THE FIRST SEA-GOING DIESEL ENGINED VESSEL.) TWO 8-CYLINDER, 4-CYCLE, DIESEL ENGINES, EACH OF 1,250 B.H.P., R.P.M., 140.



THE "RECHAD V,"—THE FIRST OF TURKISH DREADNOUGHTS.

This vessel is one of the two which are being constructed in England by Messrs. Armstrong and Messrs. Vickers. Their displacement is about 23,000 tons, and they will carry a main armament of ten 13.5-inch guns, with sixteen 6-inch quick firing guns. They have turbine engines of 31,000 H.P., and the designed speed is 21 knots.

JAPAN.

THE JAPANESE BATTLE CRUISER *KONGO*.

The Japanese cruiser *Kongo*, launched at Barrow-in-Furness (Messrs. Vickers, Limited), has naturally attracted considerable interest, because it was recognized that the wealth of experience gained in the recent war afforded valuable guidance to the Japanese authorities in stating the requirements to be met from the point of view of fighting efficiency.

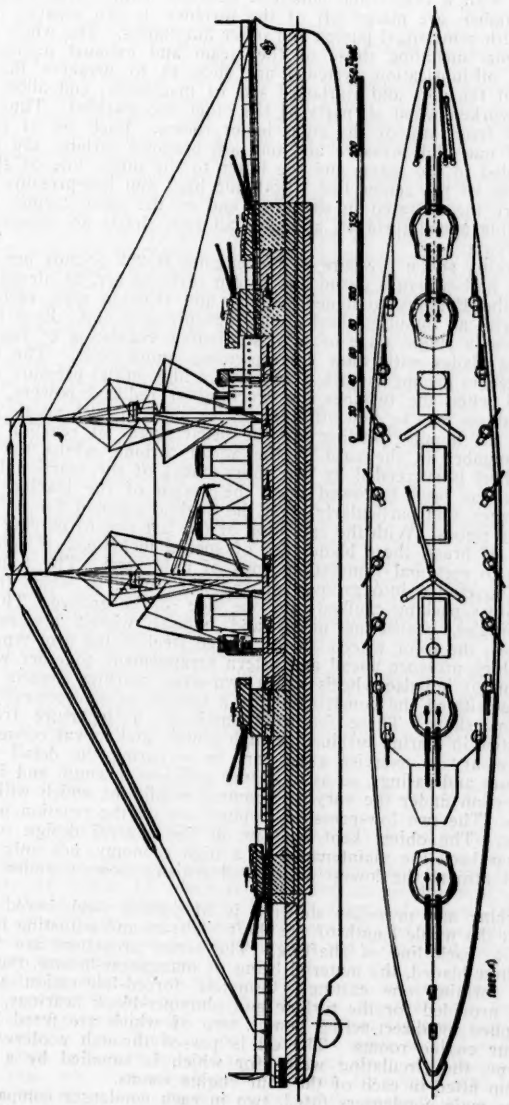
The principal dimensions are as follows: Length, 704 feet; breadth, 92 feet; draught, 27 feet 6 inches; displacement in tons, 27,500; speed, at least, 28 knots; full coal capacity, 4,000 tons; oil fuel capacity, 1,000 tons. Shaft horsepower approximating to 70,000.

Armament: Eight 14-inch, sixteen 6-inch, and sixteen smaller guns, with a large number of submerged broadside torpedo-tubes.

The eight 14-inch guns are mounted in pairs in four barbettes, two of which are located forward and two aft, all on the center line. These barbettes are arranged, and the elevation of the guns is fixed, so that four may fire forward and four aft, while all eight may fire on either broadside. The guns to the rear of the bow barbette are at a higher elevation, and in the case of the two after barbettes the forward pair are at a higher elevation, although, owing to the relative positions of the one to the other, the difference in height is not so great as in the forward guns; but, owing to the absence of any obstruction abaft the guns, it is possible to fire all four guns astern in line with the keel. It is important to note that the Japanese authorities have considered it necessary to arrange that four guns should be fired astern. An important development was also introduced in connection with the 6-inch guns, which are of 50 calibers in length. These are fitted in casemates on the upper deck level. The advantage achieved in the design of these casemates is that the guns can be trained through a wide arc. This general policy is consistent with the arrangement made since the war by the Japanese authorities, who have in all cases adopted, in addition to the primary guns, a large battery of 6-inch weapons, a quality which has since become general amongst all naval Powers. The sixteen smaller guns are placed in convenient positions on the superstructure, in order to command a high elevation.

Notwithstanding the very powerful armament provided, the armored protection is most effective, particularly against torpedo attack. Indeed, more weight has been allocated to this than in most ships. The main broadside armor is of special-quality steel, and is equal in thickness to that of any battle cruiser yet designed, and is carried to the height of the boat deck, which is continued on the same level as the forecastle, forming a gun citadel, into which the 6-inch gun casemates are worked. The main belt extends considerably below the water line, and under this, again, there is an auxiliary armor belt extending some distance below the normal armor shelf. There is a special arrangement of armored bulkheads protecting the vital parts of the ship; the magazines, for instance, being completely surrounded with special steel armor. There is an armored deck at the water-line level, and in addition to this there is an armored deck closing in the ship from stem to stern at the level of the top of the side armor.

In the case of the propelling machinery, the same idea of adopting the best practice of the moment has been carried into effect; the water-tube boilers burn oil fuel as well as coal; the turbines are of the combined impulse and reaction type; and in the details and in auxiliary engines several interesting improvements have been effected. To ensure safety, the boilers are arranged in eight compartments, four on each side of a center-line bulkhead, which extends throughout their whole length, while the coal bunkers are also disposed to afford protection. Again, the engines—two sets of turbines on four shafts—are arranged in two com-



THE IMPERIAL JAPANESE BATTLE-CRUISER "KONGO."

partments, with a center-line bulkhead between them. The condensers, four in number, are placed aft of the turbines in two separate rooms, together with centrifugal pumps and other auxiliaries. The whole of the arrangements, including those of the steam and exhaust piping, feed, drain and oil-lubrication systems, are such as to preserve the independence of the port and starboard sets of machinery, and allow either set to be worked when all parts of the other are disabled. Thus steam can be got from any of the eight boiler rooms. Each set of turbines consists of one high-pressure and one low-pressure turbine, the former being coupled to the outer, and the latter to the inner, line of shafting on each side of the center line. Both the high and low-pressure astern turbines are incorporated in the after end of the same casings as the corresponding ahead turbines, and thus all four shafts are available for astern working.

The working steam pressure at the engines is 205 pounds per square inch. The high-pressure ahead and astern turbines are, as already indicated, of the Parsons combined impulse and reaction type, each being provided with an impulse wheel at the high-pressure end. Each impulse wheel carries a single stage of impulse blading, consisting of four rows of rotating blades with their corresponding guide-vanes. The nozzles are arranged in groups in such a way that a high initial pressure may be maintained when the turbines are working at reduced powers, as the supply of steam may be shut off from one or more of such nozzle groups. Following the impulse blading are the usual stages of reaction blades, seven in number in the case of the ahead turbine, whilst the astern impulse wheel is succeeded by two short stages of the reaction blading. Great care has been bestowed upon the design of the blading of the impulse stages, and particularly upon the method adopted for its attachment to the rotors. With the exception of the last row of moving blades, which are of brass, these blades are of special nickel-coated mild steel, welded on to sectional foundation rings of mild steel. The foundation rings are dovetailed into grooves on the wheel rim, to which they are fixed by brass packing caulked into one side of the groove. The rotor drums, spindles, connecting pieces and impulse wheels are made of forged steel; the rotor wheels are of forged steel of the arm type. The combined high-pressure ahead and astern arrangement, together with the introduction of impulse wheels and a two-stage cruising element in one casing, necessitated the construction of a turbine of exceptionally large overall dimensions. These features constituted a departure from the usual practice in marine turbines of high power, and it was consequently found necessary to exercise great care in preparing the detail designs of the rotors and casings, so as to ensure sufficient strength and freedom from distortion under the varying steaming conditions which will obtain on service. The two low-pressure turbines are on the reaction principle throughout. The object kept in view in the general design of these turbines has been the maintenance of a high economy, not only at full power, but also at the lowest anticipated cruising powers under service conditions.

The turbine and propeller shafting is of forged steel, bored hollow throughout the whole length of the shaft. Thrust and adjusting bearings are fitted to each line of shafting. The screw propellers are four in number, three-bladed, the material being of manganese-bronze, the blades and boss forming one casting. Complete forced-lubrication arrangements are provided for the turbine and plummer-block bearings, the oil being supplied by direct-acting pumps, two of which are fitted in each of the four engine rooms. The oil is passed through coolers of the tubular type, the circulating water for which is supplied by a direct-acting pump fitted in each of the four engine rooms.

The four main condensers fitted, two in each condenser compartment, are of the "Uniflux" type, with the tube casings built up of steel plates and angles. The end covers are of cast iron, to protect the tubes. &c.,

from galvanic action. There are two independent air pumps for each pair of main condensers, the pumps being of the "Dual" type, each having one air and one water barrel. The circulating pumps for the main condensers are of the centrifugal type, two for each pair of main condensers. The pumps are driven by independent two-crank engines fitted with forced lubrication. Two auxiliary condensers, one in each condenser compartment, take the exhaust steam from the auxiliary machinery. In connection with the auxiliary condensers, two air and two circulating pumps are provided.

In the boiler room there are eight main and eight auxiliary feed engines of the direct-acting type; the capacity of the auxiliary feed service is equal to that of the main. There are four grease extractors in connection with the delivery pipes of the main air pumps and two for the auxiliary air-pump discharges. Fourteen steam pumps are fitted for fire and bilge purposes and for emptying the double bottoms of the compartments. Four of these pumps are fitted in the engine rooms and the remainder in the boiler compartment; those fitted in the boiler rooms are also capable of being used in connection with the See ash ejectors, ten of which are provided.

The thirty-six water-tube boilers in the ship are of the Yarrow large-tube type, arranged to pass the products of combustion into three funnels. They are loaded to 275 pounds pressure per square inch, and are arranged to work under forced draft with closed stokeholds. The boilers are arranged to burn oil fuel as well as coal; a complete installation of pumps, heaters, filters and collectors, with all connections, being provided and fitted complete for this purpose. Electrical indicators for regulating the firing of the boilers are fitted complete in the boiler room, the furnaces being numbered to correspond with numbers which are periodically displayed on the indicator dials. Five steam-driven air compressors are fitted for cleaning the boiler tubes externally by air jet.

The fans for supplying forced draft to the boilers number thirty-four—thirty-two of the single-breasted and two of the double-breasted type. They are driven by double-acting steam engines fitted with forced lubrication. The six evaporators are worked with steam taken from the auxiliary exhaust system or the auxiliary steam service. The two distilling condensers are of the cylindrical type, capable of condensing all the steam from any four of the evaporators. All the evaporators are connected so that any of them can deliver their vapor either to the distiller or the auxiliary condenser in their own compartment.—"Engineering."

THE JAPANESE BATTLESHIP *KAWACHI.*

CONSTRUCTED AT THE IMPERIAL DOCKYARD AT KURE.

The battleships *Kawachi* (now in commission) and *Settsu* (completing for sea), the former of which is illustrated, as are the first all-big-gun ships of the single-caliber type to be built for the Japanese Navy. Laid down in 1909 at Kure and Yokosuka respectively, the ships will have been well over three years under construction, the *Settsu* not being due for completion until this summer.

On a displacement of 20,750 tons they carry an armament of twelve 12-inch, ten 6-inch, and twelve 4.7-inch guns, together with five torpedo tubes. Their original design gave them fourteen big guns, the axial turrets containing three guns apiece; but these triple turrets were replaced by the usual twin type at an early stage of their construction, and the six positions are disposed after the German *Helgoland* method—that is, one fore and aft and one at each corner of the superstructure. The ships are thus enlarged and modified *Katoris*, with two 12-inch, replacing the

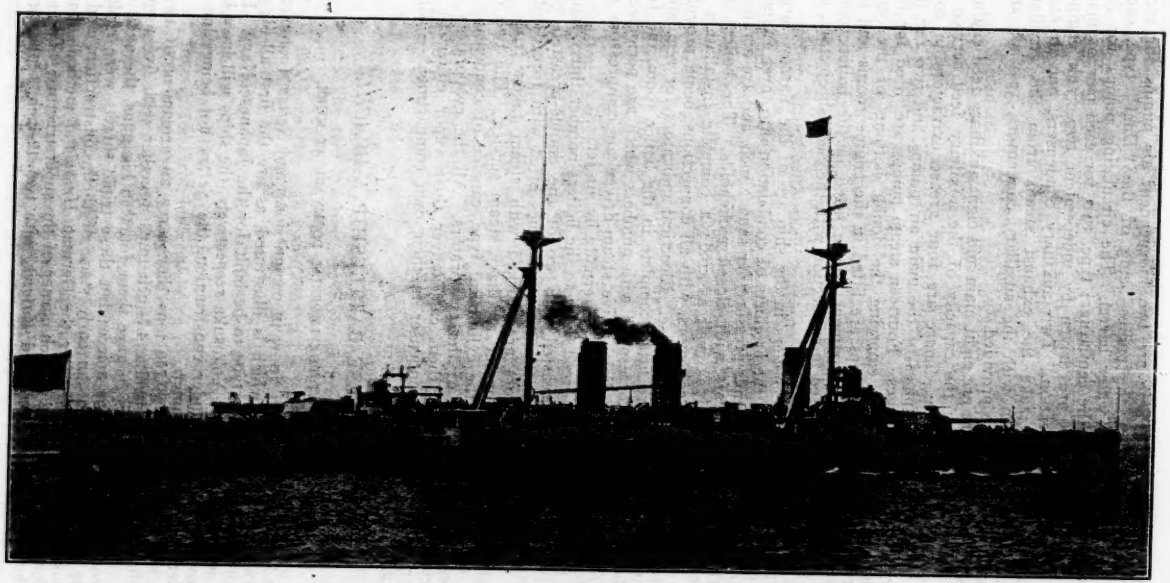
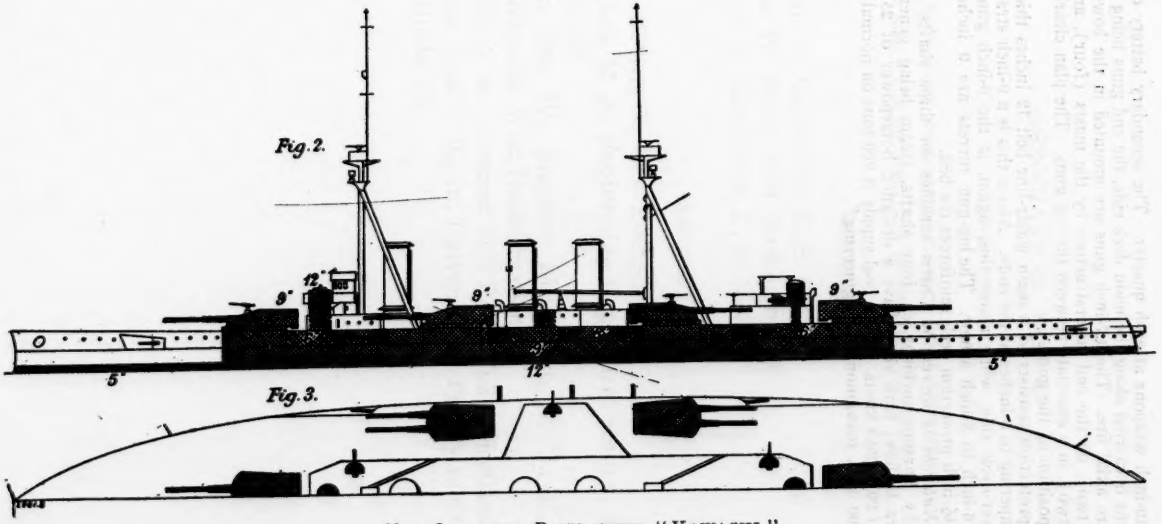


FIG. 1.—THE JAPANESE BATTLESHIP "KAWACHI."



THE JAPANESE BATTLESHIP "KAWACHI."

single 10-inch weapons at each quarter. The secondary battery of ten 6-inch guns is carried along the main deck side, the end guns being recessed to secure axial fire. The 4.7-inch guns are mounted in the bows (two), stern (two), on the end superstructure by the masts (four), and amidships (two), in a casemate between the big guns. The plan clearly shows the disposition of the guns.

The protection consists of a main water-line belt 12 inches thick amidships, tapering to 5 inches at the ends. Above this is a 9-inch strake along the lower-deck side, while above this, again, is the 6-inch gun battery behind 6-inch to 8-inch armor. The big-gun turrets are 9 inches thick, and a 2½-inch protection deck reinforces the belt.

The *Kawachi* is driven by Curtis turbines on three shafts, and the *Settsu* by Parsons turbines on four shafts, steam being generated by Mijabara boilers; both ships have a designed horsepower of 25,500 and speed of 20 to 20.5 knots. Their coal supply is 900 tons on normal draught and 2,500 tons maximum.—"Engineering."

ASSOCIATION NOTES.

The following new members and associates have joined the Society since the publication of the last JOURNAL:

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Duncan, W. Butler, Port Washington, N. Y.
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Mitchell, A. R., Kelwood, Mill Road, Yoker, Scotland.
Stannard, Prof. J. H., Jr., University of Pennsylvania, Philadelphia, Pa.